ASHRAE
OFFICIAL PUBLICATION

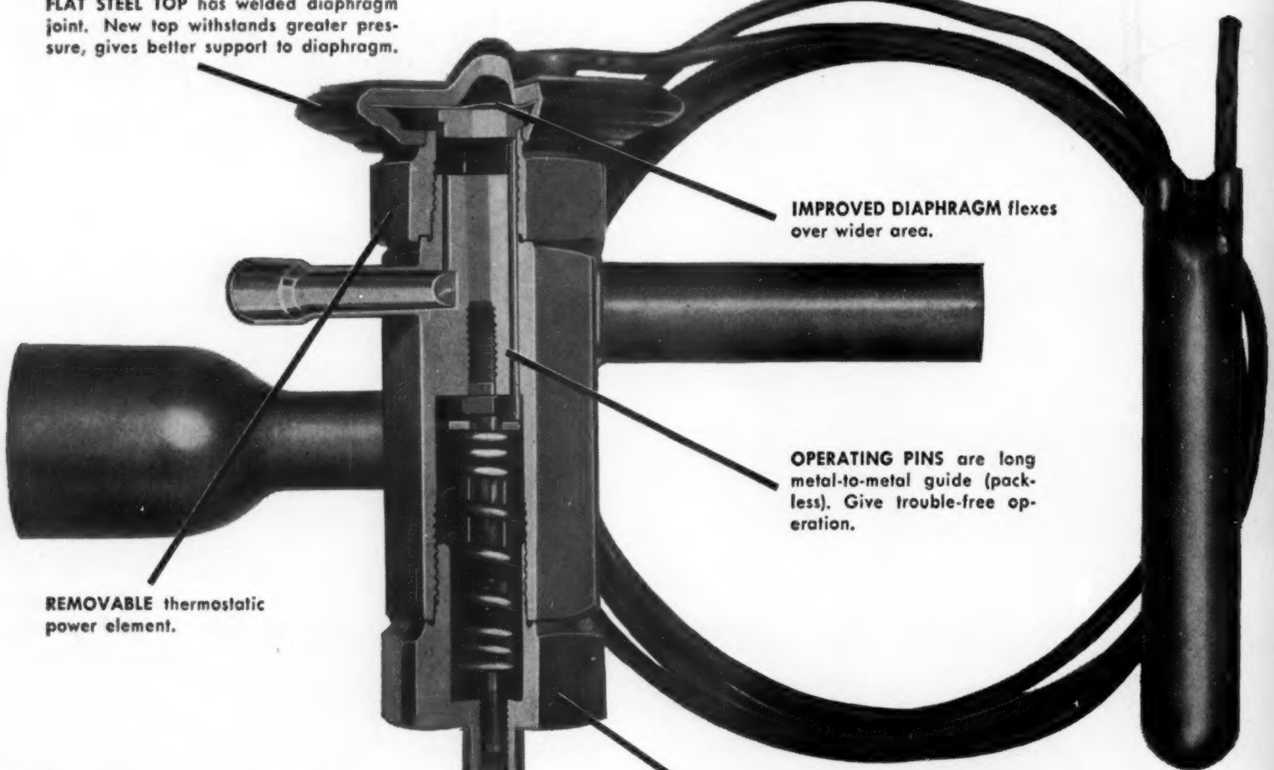
JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS

NOVEMBER 1961

"Flat-top" valve with improved diaphragm withstands over one million normal life cycles



New AP model 214 "flat-top" thermostatic expansion valve

This new, improved thermostatic expansion valve was built to last and last. In fact, the new model 214 "flat-top" has a life span of over 1,000,000 normal life cycles. It's the ideal valve for trouble-free replacement on large commercial applications.

Just the valve for heat pump applications, too! Sudden withdrawal of high pressure on the outlet side coupled with high thermal bulb pressure was causing tremendous strain on the diaphragm. Result: Life expectancy on some valves used on heat pump applications was as low as 7000 life cycles. CC engineers extended valve longevity with the improvements shown above — features that enable the new "flat-top" to withstand over *100,000* life cycles of extreme pressure fluctuations incurred during heat pump reversals.

The new A-P model 214 is part of industry's most complete line of refrigeration and air-conditioning controls. Take advantage of Controls Company of America's single-source availability — your assurance of dependable performance. Write today for complete details.

Visit Booth 803 at Heating and Air Conditioning Exposition, Chicago, Feb. 13-16, 1961.

Creative controls for industry

CONTROLS COMPANY OF AMERICA

HEATING AND AIR CONDITIONING DIVISION

2456 N. 32nd Street, Milwaukee 10, Wisconsin • Cooksville, Ontario • Zug, Switzerland

NOVEMBER 1961

VOL. 3 NO. 11

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Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

FEATURE

NEWS

DEPARTMENTS

Member
Audit Bureau of Circulations

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Letters to the Editor

AS TO BOILER EFFICIENCY

To the Editor:

I have read with considerable interest the article by A. Eugene Congress in the September 1961 issue of the JOURNAL "Integrated Technique for Estimating Annual Energy Use of Central Air Conditioning Plants." Author Congress is certainly to be congratulated for his detailed analysis of energy consumption of the various types of systems presented in the paper for this is one of the finest comparisons I have ever seen. An analysis of this nature is needed definitely when comparing the economics of an air conditioning system.

However, there are two items in the analysis which do an injustice to the all electric system when compared to the other systems discussed in the paper.

First and highly important is the boiler efficiency used by Mr. Congress in making the analysis. There is no area in which there is more loose thinking than the subject of boiler efficiency when it is used for the prediction of yearly fuel consumption. Many consulting engineers and architects quote all sorts of authorities and manufacturers' guarantees. The term "boiler efficiency" is used in a number of different senses but for this analysis the only correct efficiency to use is the "seasonal efficiency."

There is a significant difference between the test efficiency and the seasonal efficiency of a boiler plant. The test efficiency is obtained under strictly controlled conditions. Such tests are usually run in the following manner: The boiler is cleaned on both the fire side and the water side and run for at least 24 hr on the fuel used. The boiler is put on the pre-determined load, usually its rated capacity, for several hours before readings are taken. During this tune up period, the air/fuel ratio is adjusted to give maximum efficiency. During the test, the load is held constant and the operation is strictly controlled to obtain maximum efficiency. Under these conditions there is no problem to get design efficiencies of 75 to 80% for any fuel and load conditions.

No heating or industrial boiler ever runs under such strictly controlled conditions for there are undeterminable amounts of heat lost. As the boiler becomes older its efficiency is further reduced so that it is not realistic to use boiler efficiencies of 80% particularly over the entire 15-yr period under consideration.

My second item is the present worth factor at 4% annual compound interest. As stated in the paper, this is the rate the government pays for long-term money.

The present worth method usually involves an assumption that the machines or structures being composed are renewed at regular intervals at the same prices which prevail at the time of the economy study. In any case, it is necessary to make some assumption as to the recurrent investment costs if present worth methods are used; if the estimator is unwilling to make a specific forecast of change which is to be used in his economy study, he is forced to assume a continuance of present conditions.

It should be realized that one objective of interest calculations in economy studies is to insure that investments should not be made which do not show the prospect of a return. Estimators who use present worth or capitalized cost comparisons commonly use as an interest rate, the bare cost of borrowed money, rather than the rate of return required to justify an investment; thus their cost comparisons frequently have an undetected bias in favor of the alternative involving higher first cost.

The interest rate used for present worth conversions in economy studies should be a high one; it should be the rate of return required to justify an added investment. In this case it should have been 10 to 12%.

Were these two items adjusted in the comparisons provided by the author, the paper might take quite a different form.

P. E. CHANEY
Commercial & Industrial Division
Texas Electric Service Co.
Fort Worth, Texas

DOES NOT DIFFER APPRECIABLY

To the Editor:

Our Navy experience confirms the position that it is not the boiler test efficiency but rather the operating or seasonal efficiency that counts. Most heating plants operate at partial load during the months when air conditioning is required and at the lower end of the boiler efficiency curve. By the use of the true operating efficiency as a starting point, the increase in the steam load to supply air conditioning requirements is accomplished without a corresponding increase in many of the boiler losses. In the instance cited in the paper, the boilers are equipped with heat recovery equipment. The efficiency range of 78 to 82% noted in the paper is a year-in and year-out operating efficiency. The summer efficiency of 80% assumed in the paper (see discussion under "Incremental Boiler Efficiency") is 5.5% less than the actual boiler test efficiency. However, it is the incremental operating boiler efficiency of the extra summer load that really must be considered in estimating the amount of fuel that will be consumed for the incremental steam supply for the air conditioning season. An example of this computation is given in the paper; but instead of using the 86.4% incremental efficiency derived, 80% is used—the remaining 6.4% being allotted to the extra power consumed by boiler auxiliaries. This, it must be conceded, is not an ungenerous allowance.

We cannot concur with Mr. Chaney that a boiler's age alone affects its efficiency. A new boiler, if not properly operated and maintained, can show a

(Continued on page 8)

BULLETINS

Interception Filters. Design consideration, typical performance data and features of the Smokestop line of disposable interception filters are included in Flyer 171. Applicability of the units is for room recirculators, kitchen range hoods and other applications requiring controlled, positive filtration.
Cambridge Filter Products Corporation, 738 E. Erie Blvd., Syracuse 1, New York.

Squirrel Cage Motors. Integral hp (one through five), polyphase, squirrel cage motors are presented in Flyer 500. General and special purpose, open drip-proof, the motors are available with a variety of mountings, electrical and mechanical modifications for all standard voltages, frequencies and speeds. Typical performance curves and Nema frame assignments by hp and speeds are included.
Leland Ohio Electric Company, Sub of Howell Electric Motors Company, 16316 W. Seven Mile Rd., Detroit 35, Mich.

Coolers and Condensers. Extensive information on horizontal core, air-cooled heat exchangers for cooling and temperature control of oil, water and gases or for condensing vapors is provided by 16-page Catalog 561. Illustrations of typical applications are presented and structural features of various cooling sections and available cores are illustrated and described. Design features are covered, accessory equipment is listed, various methods of cooling using HC units are detailed in diagrams and required information for recommending HC units and a unit nozzle selection chart are included. Dimensional drawings and specification data complete the catalog.
Young Radiator Company, Racine, Wisc.

Dispersives. Formulated to minimize adherent sludge formation in boilers and recirculating cooling systems, dispersives are discussed in four-page Bulletin HSP-941. Selection of dispersives based on results of physical and chemical examinations of deposits and samples needed to conduct the examinations are covered. Several case histories dealing with sludge accumulation problems in various boilers and cooling systems also are described.
Hagan Chemicals & Controls, Inc., Hagan Ctr., Pittsburgh 30, Pa.

poor efficiency. But a clean, tight boiler, free of scale and with controls in proper adjustment, can maintain the same high operating efficiency throughout the years. Operation and maintenance of the "old gal," not her age, determines her performance. This is not intended to overlook the advantage of retiring an old boiler when savings from operation of a later, more efficient boiler can amortize its investment within a reasonable period of time.

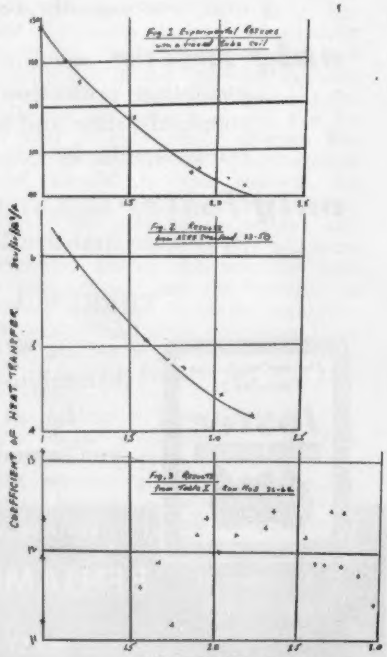
In regard to Mr. Chaney's comment on "present worth," the author is concerned only with the evaluation of the present worth of the difference in operating cost of alternative systems. The study of the paper shows that the yearly difference between the highest operating cost scheme and the lowest is $5100. The question to be answered is whether the conversion of this difference to present capital worth is correct. To capitalize an annual return (or savings) of a given amount of money over a period of time is not a highly technical problem. A rate of return of $11.118 per year for 15 years will totally retire a bond of $100 at 4% interest. Therefore the present worth of these returns (or savings) is $100. On the same basis, the savings shown of $5100 per year will retire an immediate current investment of $56,700 at 4% interest. This is the present worth of the indicated annual savings. The 11.118% (11.118 "present worth" factor) does not differ appreciably from the figure of 10 to 12% which Mr. Chaney indicates is the required return to justify investment.

A. EUGENE CONGRESS
Dept. of the Navy

TO THE OPPOSITE CONCLUSION

To the Editor:

In W. L. Bryan's article in the September issue of your Journal it was found that "the coefficient of heat transfer for dry cooling was lower than when the coil was operating with dehumidification." This result was apparently expected "since the air turbulence is greater during dehumidification." A similar conclusion is also contained in Reference No. 9 where



the author L. C. Bull states, "dehumidification has the effect of reducing the thermal resistance between air and external coil surface."

Prof. Bryan's work prompted me to examine some test results obtained in the laboratories of F. H. Biddle Ltd., with a finned coil on chilled water at different humidities. These results appear to lead to the opposite conclusion. The tests were done largely to the ASRE Standard 33-58 in connection with a specific inquiry. The tests were done at a constant value of air velocity and water velocity, but at various values of humidity. The sensible heat transferred was divided by the log-mean temperature difference between entering air/leaving water, and leaving air/entering water, by the face area of the coil in sq ft, and by the number of rows. The resulting heat transfer coefficients are plotted on a curve (Fig. 1) against the coil multiplier M, defined as the ratio of overall total to sensible heat removal.

It may be seen from this curve that the coefficient of heat transfer apparently decreases with increasing M, i.e. with increasing dehumidification. The coefficient with a dry coil, i.e. at M = 1, is higher than the coefficient with a wet coil at any degree of wetness. The suggested physical explanation for this is that with increasing dehumidification, more of the coil surface is covered with a film of water, (or the film increases in depth), and that this film being a relatively poor heat conductor acts as a heat barrier, and decreases the rate of heat transference.

By taking results from Fig. A of the ASRE Standard, a similar curve can be plotted. This has been done at an air velocity of 400 fpm, water velocity 1 fps, the coefficient in this case being expressed per sq ft of external surface area. The curve is shown in Fig. 2, and it will be seen that in this case, too, the coefficient of heat transfer appears to decrease with increasing dehumidification.

It is also possible to express the results of Table I in Prof. Bryan's article in a similar fashion. Test run numbers 36 to 62, which have been done at more or less the same air velocity, have been calculated to obtain the M values, and the results have been plotted in Fig. 3. It will be seen that the scatter of the results is high, and one would hardly be justified in drawing any trend line. Certainly there is no indication of the coefficient increasing with increasing dehumidification; at best, it appears as though the coefficient is more or less constant, whether the coil is wet or dry. Perhaps this is not surprising, since the coil is made from bare tubes, and probably the condensed water does not remain on them to the same extent as it does on a finned coil. Hence, there would be less interference to the transfer of heat from the air.

R. W. FEARN
Chief Physicist

F. H. Biddle, Ltd.
Hammersmith,
London, England

AUTHOR BRYAN SAYS ITS PREMATURE TO CONCLUDE

To the Editor:

At this time, I can neither fully agree nor strongly disagree with correspondent R. H. Fearn since the work on a fin tube water coil has not been completed. Some points based on the data collected on the bare tube coil operated at substantially constant refrigerant temperature follow.

The reported data gave higher coefficients of heat transfer when both heat and mass were transferred than when heat alone was transferred. The heat transfer coefficient was fundamentally a function of the air velocity through the coil and little if any change in the coefficient as a function of degree of dehumidification was evident. The scatter of points of Mr. Fearn's Fig. 3 is due to the differences in air velocity. There is a difference between the change of the heat transfer coefficient with air velocity for the dry and for the wet coil (Paper Fig. 4). This difference is reduced as the air velocity and thus relative degree of turbulence is increased.

The conservative estimate of the temperature drop across the water film on the bare tubes was 0.4 F which was neglected. The coefficient of heat transfer, based on the air dry bulb to tube surface temperature difference, would be slightly higher if the water film temperature drop were important and considered.

The temperature potential was known to be the Log-mean temperature difference with a constant refrigerant temperature. For the bare coil the surface temperature was measured experimentally and thus the heat transfer coefficient was completely defined. In the case of a water coil where the refrigerant temperature varies in counter-crossflow the proper temperature difference and thus the coefficient of heat transfer is not satisfactorily defined. Conclusions on such water coils must be considered on the basis of the methods used for calculating the coefficients.

W. L. BRYAN
Engineering Division
Case Institute of Technology

WHO'S WHO IN ASHRAE

Insofar as possible these listings will each appear twice a year

ASHRAE OFFICERS, DIRECTORS COMMITTEES, STAFF

See page 80, this issue

REGION AND CHAPTER OFFICERS

See page 86, October JOURNAL

RESEARCH AND TECHNICAL COMMITTEES

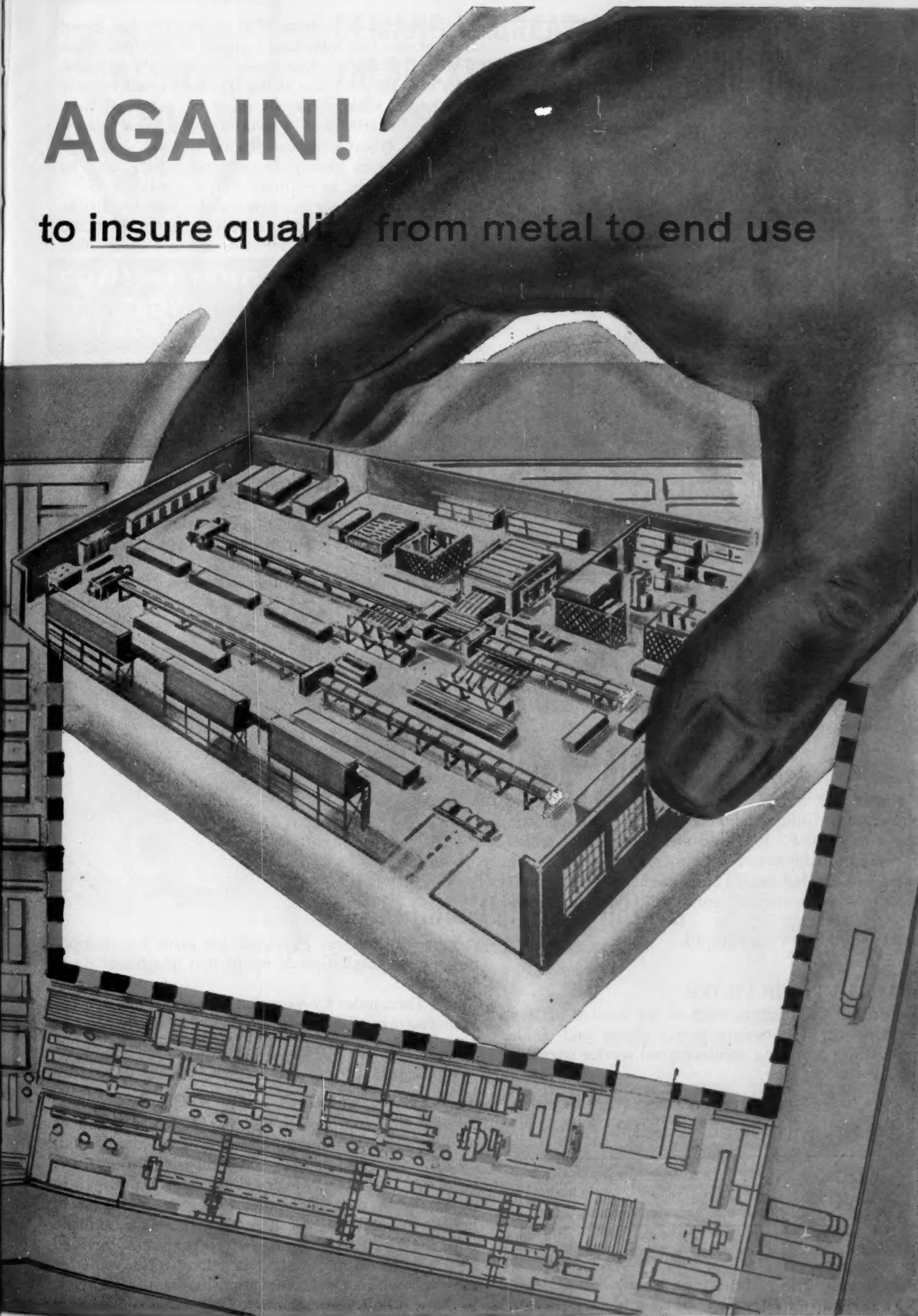
See page 76, September JOURNAL

STANDARDS PROJECTS

See page 84, May JOURNAL

INTER SOCIETY COMMITTEES

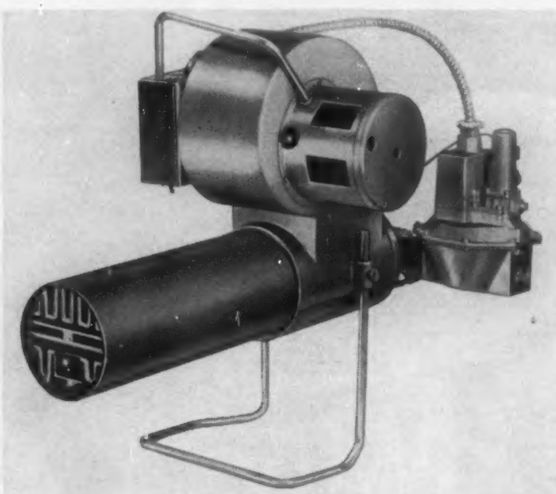
See page 88, this issue


AGAIN!
to insure qual
from metal to end use

PARTS AND PRODUCTS

POWER GAS BURNER

With addition of Model 800 to the Economite line, these burners now cover a capacity range of from



75,000 to 800,000 Btu/hr. Factory-tested on gas, the new unit has a 5-in. diam tube, is packaged and ready for installation, and is available with either thermoelectric or electronic controls.

Mid-Continent Metal Products Company, 2717 N. Greenview Ave., Chicago 14, Ill.

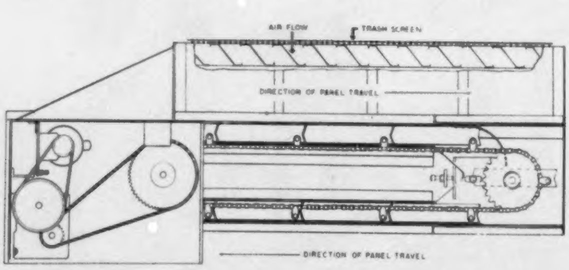
UTILITY CABINETS

Available in 5 and 13-cu ft capacities, this line of low temperature utility cabinets has five temperature ranges to –120 F. Featured are hermetically sealed, air-cooled, halogenated hydrocarbon refrigeration systems. Included on the instrument panel are built-in, adjustable indicating controller, pilot lights and selector switches.

Deltair Products, Villa Park, Ill.

HEAVY DUTY AIR FILTER

For use on installations such as air intakes of large gas turbines in stationary power plants and similar applications requiring uninterrupted service over long



periods with little maintenance, a new line of Model HD automatic air filters has been introduced. In rated maximum capacities from 3720 to 130,500 cfm, based on a net effective filter face velocity of 450 fpm, these oil-wetted filters are guaranteed against oil carryover. Units may be used also at the standard rated velocity of 505 fpm, with maximum rated capacities of from 4166 to 146,160 cfm. Powered by a heavy duty, polyphase, 60-cycle motor driving through a double reduction worm gear reducer, these filters may be adjusted to cycle, as required by the application.

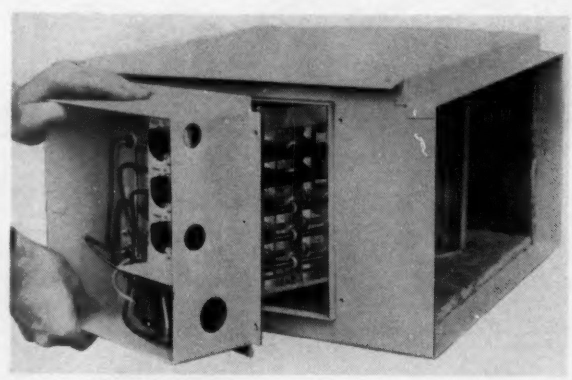
From one to four sections may be powered by the single-unit drive, by means of a through shaft and self-aligning couplings between the adjoining filter sections. A shear pin protects the drive against sudden overloads. Accessories include adhesive oil filtering systems, automatic sprinkler systems and adhesive oil reservoir heaters.

Rockwell-Standard Corporation, Air-Maze Div, 25000 Miles Rd., Cleveland 28, Ohio.

ELECTRIC DUCT HEATERS

Designed for use as supplemental heaters in heat pumps, as air heaters in ventilating systems, or in conjunction with straight air conditioning units, these easily installed duct heaters are available in six, eight and twelve-kw units for single or three-phase, 240-volt service; 208 or 480-volt elements are available on special order.

Each duct heater includes a main contactor, combination automatic reset limit and duct control, low voltage terminal strip and a plate-mounted control



voltage transformer. Provisions are made for multiple staging by installation of up to two additional ductstats.

Norris-Thermador Corporation, Thermador Electrical Manufacturing Company Div, 5119 District Blvd., Los Angeles 22, Calif.

RELATIVE HUMIDITY INDICATOR

Portable and compact, this light-weight psychrometer is designed for rapid determination of relative humidity in general commercial and industrial applications. Readings of wet and dry bulb can be taken in proximity to critical manufacturing processes or in hard-to-get-at locations. Measurements may be made with the instrument in any desired position. Provided in the water supply system is a long term, refillable reservoir which carries moisture, by means of a wick, to the wet bulb thermometer. Two standard flashlight

HENRY VALVE CO.

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DEMAND IS GREAT . . . EVERYONE IS BUYING THESE EXTRA VALUE PRODUCTS

"DRI-VUE" Moisture-Liquid Indicators	"DRI-COR" Filter-Driers Molded Core	"GOLDEN BANTAM" Diaphragm Packless Valves

"DRI-VUE" INDICATOR PORT

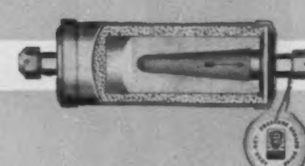
Actually Spells "WET" or "DRY" with element Color Variation—Largest Visible Element Area.

"DRI-VUE" INDICATORS

Double Port Moisture-Liquid Indicators, Large O.D.S. Sizes. Triple Sealed Ports.

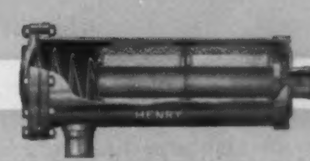
"DRI-COR" FILTER-DRIERS

Two Stage Drying; Molded Core and Granular Desiccant. Abso-dry Processed.



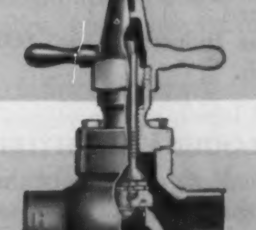
"DRI-COR" Cartridge DRIERS

Molded Filter Core and Granular Desiccant. Split Second Cartridge Installation.



SHUT-OFF VALVES

Packless and Packed with Flanged and Integral Connections. Wide Range of Sizes and Types.



Only Henry offers you this complete variety of types and sizes, from 1/4" flare packless to 4 1/8" O.D.S. packed Wing Cap valves, from 1 ton sealed type to 165 ton cartridge type filter-driers and 1/4" flare single port to 2 1/8" O.D.S. double port moisture indicators.

HENRY VALVE COMPANY

For Refrigeration, Air Conditioning and Industrial Applications

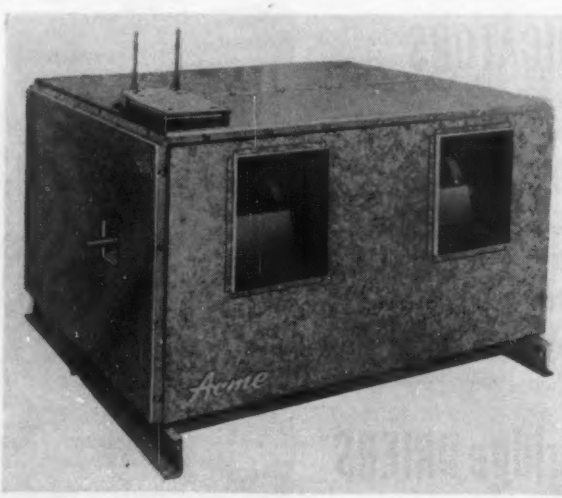
MELROSE PARK, ILLINOIS, U.S.A. CABLE: HEVALCO, MELROSE PARK, ILL.

batteries power a fan of low current drain to provide proper air circulation and evaporation. Affixed to the back side of the case is a comprehensive chart which provides the percentage of relative humidity.
Cybernetics, Inc., 136 Washington St., Paterson 1, New Jersey.

AIR HANDLERS

To heat or cool, remove dirt, humidify or dehumidify, or serve as simple air circulators, these units will operate with either steam or hot water as a heat source and with either a water chiller or direct expansion coil as a cooling source. Incorporating a number of basic improvements, the new line comprises a total of 37 horizontal, vertical and multi-zone type units in 14 sizes ranging in capacity from 600 to 44,000 cfm and providing a nominal outlet velocity of 1600 fpm. Included in the line are 14 horizontal units (one of which is shown) ranging in size from 2.7 to 61 sq ft, 12 vertical units from 2.7 to 41 sq ft and 11 multi-zone units, with up to 19 individual zones available in the larger sizes.

Air handlers in the line are constructed of 14 and 16-gauge galvanized metal with 2½ oz of electro-deposited zinc per sq ft and feature 1-in. standard glass fiber insulation. No angle irons are used. From two to four coil sizes in either left or right-hand combinations are available for each size casing. Six

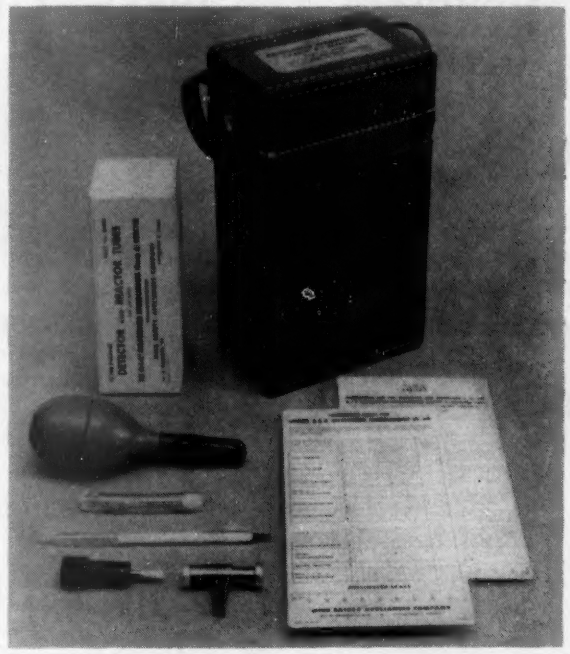


motor positions and four fan discharge positions are offered.
Acme Industries, Inc., 600 N. Mechanic St., Jackson, Mich.

HALOGENATED HYDROCARBON DETECTOR

Measurement of ten of the most frequently occurring halogenated hydrocarbons is now possible with this new detector. Portable, the instrument may be used to measure toxic concentrations of such compounds as trichloroethylene, perchloroethylene, ethyl bromide, ethyl chloride, methyl bromide and monochlorobenzene.

To achieve compactness and versatility, a two-part instrument was developed: a reactor tube and a detector tube which are joined together for testing. Contained in the reactor tube are two frangible glass ampoules of chemicals, which are crushed and mixed immediately before use and serve to oxidize the halo-



genated hydrocarbon, releasing free chlorine or bromine. Two types of detector tube are utilized, one for hydrocarbons in Group A (e.g., ethyl chloride) and one for Group B (ortho-dichlorobenzene).

In operation, a sample of air is drawn through the proper detection unit by an aspirator bulb assembly. Released halogen gas reacts with the chemical in the detector tube to form a length-of-stain indication, which is correlated with the number of aspirations to determine the toxic concentration in ppm.
Mine Safety Appliances Company, 201 N. Braddock Ave., Pittsburgh 8, Pa.

HYDRONIC LINE

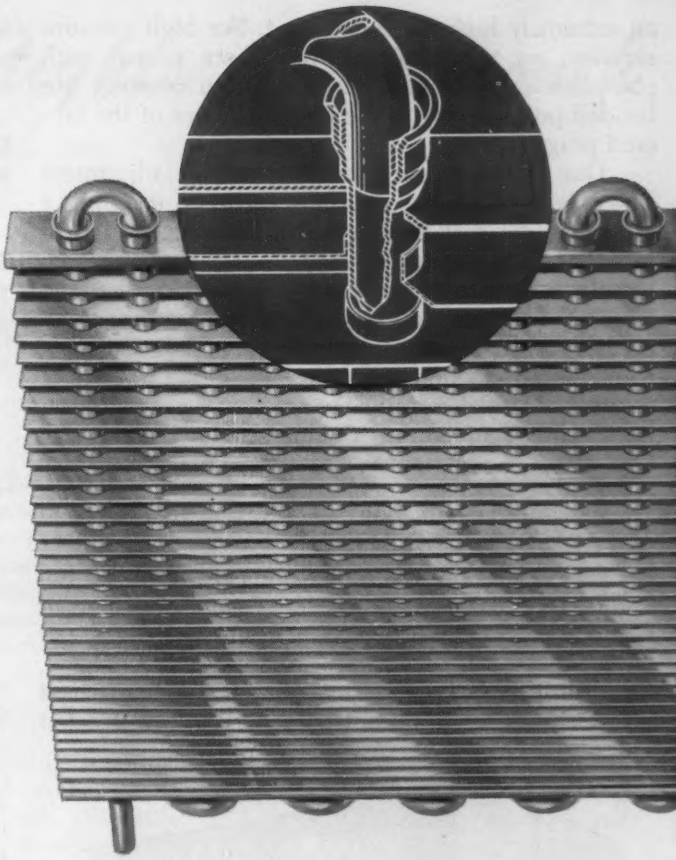
Completing this new line of cast iron, gas-fired hydronic products is the Type 26 boiler line, designed for large capacity commercial and industrial low-pressure steam or hot water applications. Offered are 29 capacities ranging from a five-section, 720,000-Btu/hr input unit to a 33-section, 5,760,000-Btu/hr input model. Boilers are rated at 180,000 Btu/hr input per flue-way. Units operate with natural, mixed, manufactured and propane gases and can be ordered for left or right placement of various controls and tankless heaters. Six sizes of tankless heaters are available in capacities ranging from 3½ to 15 gpm. Three types of storage water heaters also are offered.
Worthington Corporation, Mueller Climatrol Div, 2005 W. Oklahoma Ave., Milwaukee 1, Wisc.

TAPERED PLUG VALVES

Utilizing a high lubricity plastics coating cited as substantially reducing turning, Permaturn valves are designed to handle services ranging from jet fuels and chemicals to water and sewage. For low pressure valves (recommended for pressures to 1000 psi) the coating is Teflon, a polyfluoride thermoplastics with

Now more practical and efficient than ever!

Aluminum Tubing For Air Conditioners



Epoxy-Bonded Tube Joints Clear Way for Low-Cost Aluminum Tubing in Evaporators, Condensers

Aluminum tubing has always made a lot of design sense for fin-tube air conditioning evaporators and condensers: It's economical. It won't rust. It provides excellent heat transfer. With aluminum fins, it eliminates bimetallic corrosion.

And now, thanks to a practical tube bond method developed by Reynolds, and using epoxy resin, aluminum tubing can provide important advantages in production and performance, as well. The epoxy bond provides a strong, durable return bend—with no danger of flux corrosion.

Exhaustive tests, in the laboratory and in actual field installations, have proved the toughness of Reynolds new bonding technique—and the efficiency of Reynolds Aluminum tubing. The epoxy makes production much simpler, too. No welding, brazing, or soldering is necessary in fabricating the returns in fin-tube assemblies.

With this new bonding development, Reynolds Aluminum fin-tubing is a valuable addition to the air conditioner manufacturer's roster of cost-cutting, product-improving aluminum products: Reynolds pre-painted sheet—Colorweld T.M.—offers an efficient and attractive answer to cabinet design. Available in a full range of colors, this tough aluminum sheet has a baked-on enamel finish that takes any ordinary forming operation, and provides a handsome, durable cabinet.

Other Reynolds Aluminum products have been developed to cut weight and cost as motor mounts, fan blades, grilles, trim—even name plates.

Reynolds does not fabricate finned-tube evaporators, but does supply raw material for customer fabrication. For details, call your local Reynolds office, or write *Reynolds Metals Company, P.O. Box 2346-AP, Richmond 18, Va.*

REYNOLDS ALUMINUM

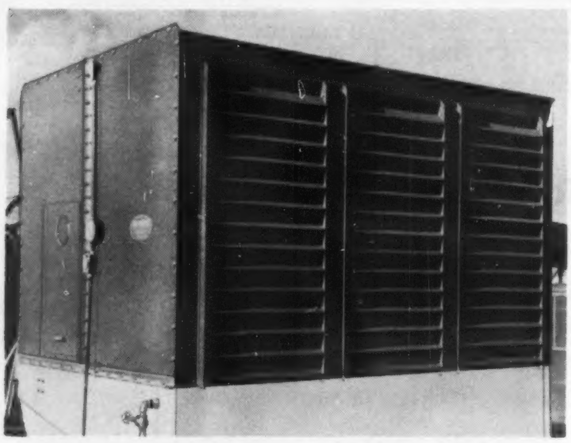
Watch Reynolds exciting TV programs on NBC: The Dick Powell **Reynolds Aluminum Show** every other Tuesday; **Say When**, weekdays; **All Star Golf**—in living color—every Saturday.

an extremely high softening point. For high pressure services, up to 10,000 psi, valves are coated with phosphate-molybdenum disulfide. Both coatings are bonded permanently to the metal surfaces of the tapered plugs.

One of two other features is a fixed adjustment assembly used on low pressure valves to maintain a constant tight seal after repeated openings and closings and to facilitate uniform turning ease. The assembly is pre-set at the factory. Second of the new features is designed for valves on extreme high pressure services and consists of a threadless stem that results in a reduction of the high operating torques of conventional high pressure valves.
Rockwell Manufacturing Company, Meter and Valve Div, 400 N. Lexington Ave., Pittsburgh 8, Pa.

AIR WASHER UNITS

Air washer-evaporative cooler units that cleanse, humidify and cool incoming air are being fabricated of rigid polyvinyl chloride (PVC). Light in weight, the units will not rot, rust, corrode or support combustion. Except for the ten-gauge steel base and stainless steel exterior framework, these air washers are constructed



entirely of PVC. The metal base is protected by PVC paint outside and a rigid PVC lining inside.

Available in three basic models to meet specific cleaning, cooling and humidifying needs, the units feature an extensive choice of spray nozzles. Cleaning and purifying of incoming air is accomplished by a spray-mist action which washes contaminants and pollutants from the air, withholding them in a settling tank. A suction strainer filters the recirculating water.

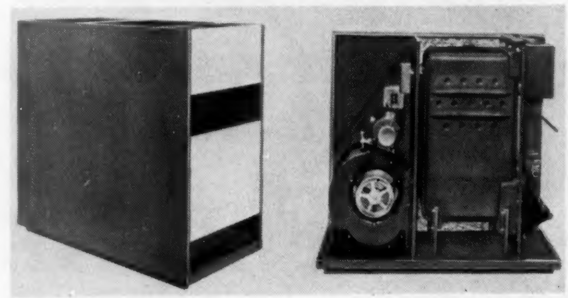
Cooling is by evaporation and occurs when the entering air strikes the spray-filled chamber, resulting in temperature reduction of both air and water. Humidification takes place when dry incoming air takes on moisture as it passes through the spray chamber.
S & C Manufacturing Company, 3533 Cardiff Ave., Cincinnati 9, Ohio.

SIX-SPEED DRIVE

Offered in 20, 25, 30 and 40-hp, a six-speed Selectro-Shift drive is engineered for use on equipment where large hp and multiple speeds are required. Hydraulic clutches activated by electronic solenoids shift the drive to any of six speeds while running under full load. Shifting is controlled by an electric push-button station which can be installed any required distance from the drive. Unit can be operated in either reverse or forward position in all six speeds.
Lima Electric Motor Company, Inc., a Sub of Consolidated Diesel Electric Corporation, Lima, Ohio.

BASEMENT WINTER AIR CONDITIONERS

Five new models of Luxaire Series SA gas-fired basement-type winter units range in capacity from 100,000



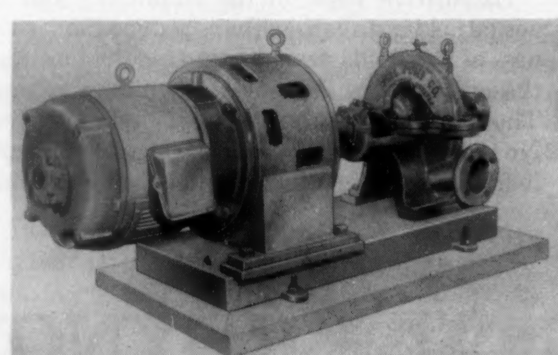
to 200,000 Btu/hr input in steps of 25,000 Btu/hr. Shipped assembled and wired (as shown at left; at right is an interior view), these units feature a heat exchanger die-formed from 16-gauge steel and end-welded with automatic submerged arc welding. A built-in draft hood is concealed inside the cabinet.
C. A. Olsen Manufacturing Company, Elyria, Ohio.

BUTTERFLY VALVES

Suitable for control of either air or gas flow up to five-psi pressure and with capacities to 20,000 cfh natural gas, manual or automatic valves are available in seven pipe sizes from one through four in. Units are of the low pressure drop type with 90-deg closing, allowing easy synchronization with shutters using suitable linkage. Ease of adjustment is permitted by a spring over-run feature. Upper and lower limits are set by stop screws regardless of motor stroke.
Mid-Continent Metal Products Company, 2717 N. Greenview Ave., Chicago 14, Ill.

PUMPING SYSTEM

For industrial, commercial and residential buildings, Ever-Flo, a constant pressure pumping system, uses a



variable speed coupling which accelerates or deceler-

(*Continued on page 26*)

News highlights of the month

TRENDS

TRRF research grants

Final contracts for The Refrigeration Research Foundation's grants-in-aid have been completed for the support of eight research projects, including studies of time-temperature tolerance on frozen foods, storage of frozen or other perishable foods, low temperature reactions on biological materials, problems of vapor movement in refrigerated warehouses and control of food poisoning organisms in food.

Lecture series

"Shock, Vibration and Acoustic Noise" will be the subject of a ten-week Tuesday evening lecture series to be held at New York University College of Engineering campus October 3 through December 12. The series, sponsored by the College of Engineering and the Division of General Education, includes such topics as "Designing for Shock and Vibration," "Instrumentation for Dynamic Measurements" and "Design of Fixtures for Shock and Vibration." Further information about the series may be obtained directly from New York University, College of Engineering, Bronx 53, N. Y.

Fuel savings

Attachment of a metal grille to any conventional high-pressure oil burner may save over 20% in a home owner's annual fuel oil bill, the Humble Oil Refining Co. announced recently. The grille breaks the fuel-air flow into swirls and currents that travel slowly enough to ignite the spray in the central region of the flame, where combustion did not occur formerly. Cost installed is $54.60.

Sampling network

Secretary of Health, Education and Welfare Abraham Ribicoff has announced the formation of a special air pollution sampling network covering Washington, D. C., and the nearby areas of Maryland and Virginia. The project will measure total oxidant, a type of air pollution associated with smog formation, produced in the atmosphere by the photochemical action of sunlight on emissions from auto exhaust and other sources. Network operation will be under the joint sponsorship of the U. S. Public Health Service and the Washington Metropolitan Regional Conference.

Air pollution

Researchers at Wayne State University College of Medicine in Detroit will study more than 4,000 experimental animals under a program designed to probe the possible effects of air pollution on human health. Special emphasis will be placed on pulmonary function, length of life, blood studies and pulmonary pathology. Among the common air pollutants which will be measured continuously, using automatic electronic instruments, are carbon monoxide, total hydrocarbons, oxides of nitrogen, total oxidants, sulfur dioxide, carbon dioxide and inorganic particulates.

Curricula research

Combining the fundamentals of labor problems, personnel management and supervision into a single course with the teaching of engineering practice is the result of a long-term curricula research project conducted by the Engineering Experiment Station, North Dakota State University. Now in its fourth term, the course was tried out for three previous terms with encouraging results.

Courtesies

ASHRAE President John Everetts, Jr., while guest of the Institution of Heating and Ventilating Engineers in London (JOURNAL, September 1961, page 89) took advantage there of the opportunity to return $18 to British Chancellor of the Exchequer Selwyn Lloyd through President Taylor of I.H.V.E. Chancellor Lloyd had sent the $18 to St. Peter's Episcopal Church in Philadelphia in personal settlement of that church's recently stated claim for $756,000 in damages and interest for a fence used for firewood by English troops in 1777.

BOOK REVIEWS

EJC annual report As issued by Engineers Joint Council, 345 East 47th Street, New York 17, N.Y., the annual EJC report, in booklet form, includes comprehensive, pertinent information about that organization. The 36-page booklet contains a list of all committees for the coming year, as well as a Directory of Board, Committees and Officer membership. 1960 activities are reported upon and Committee programs for the coming year are outlined. $1.

Automatic control Currently published by McGraw-Hill Book Company, 330 West 42nd Street, New York 36, N.Y., the Second Edition of "Automatic Control of Heating and Air Conditioning" by John E. Haines contains four new chapters dealing with electronic control circuits, electronic control units, domestic air conditioning and peripheral distribution unit control. The 374-page volume also includes over 350 illustrations. $10.50.

Evaporative cooling Theoretical principles and methods of evaporative cooling of water in ponds, spray tanks and coolers is the topic under discussion in a new 400-page volume entitled "Evaporative Cooling of Circulating Waters." Author L. D. Berman describes theoretical principles in the first section of the book, dealing with the design, types and operation of coolers in the second section. Published by the Pergamon Press, Inc., 122 East 55th Street, New York 22, N.Y. $20.

SPECIAL MEETINGS

Gas Industry Delegates to the 68th Pacific Coast Gas Association convention held in Coronado, Calif., September 13-15, heard William G. Hamilton, Jr., President of the Gas Appliance Manufacturers Association, state that the next five years will be the most important period in the history of the gas industry, due to continued public demand for even more modern design and automatic performance in appliances. Mr. Hamilton also emphasized that continuing competition and growing markets through population and increased home building are challenges to be met by the industry in the next five years.

Nema reviews progress At the 35th Annual Meeting of the National Electrical Manufacturers Association, November 16, New York, five electrical industry executives will review progress during the past year and emphasize future plans and objectives. U. V. Muscio, Executive Vice President, Fedders Corporation, will provide over 300 representatives of Nema's member companies with their first over-all view of the Nema Room Air Conditioner Section's new voluntary Certification Program. Mr. Muscio heads the Committee under whose guidance the Section will certify the cooling performance of room air conditioners produced by Nema members and other participating companies.

NOFI Exposition Committee An 18-man NOFI Exposition Committee will aid in the planning of the 24th National Oil Heat and Air Conditioning Exposition to be held in Chicago April 9-12. The 1962 Exposition will combine the Annual Show and Convention formerly sponsored by the Oil Heat Institute of America and the Promotional Conference of the National Fueloil Council, making it a comprehensive exposition of the oil heating industry.

NOTICE TO MEMBERS OF 1962 SEMIANNUAL MEETING

The next Semiannual Meeting of the American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., will convene at St. Louis, Mo., at 9:00 a.m., Monday, January 29, 1962.

Continuing through Thursday, February 1, the meeting will include four Technical Sessions covering Heating, Refrigeration and Air Conditioning. Eight Symposiums are planned covering Domestic Refrigerator Engineering, Industrial Ventilation, Food Refrigeration, Combined Heating and Cooling Systems and Room Air Conditioners. There will be Forums on immediate industry problems.

Every member who can do so should plan to attend this St. Louis Meeting.

JOHN EVERETTS, JR.
President

R. C. CROSS
Executive Secretary

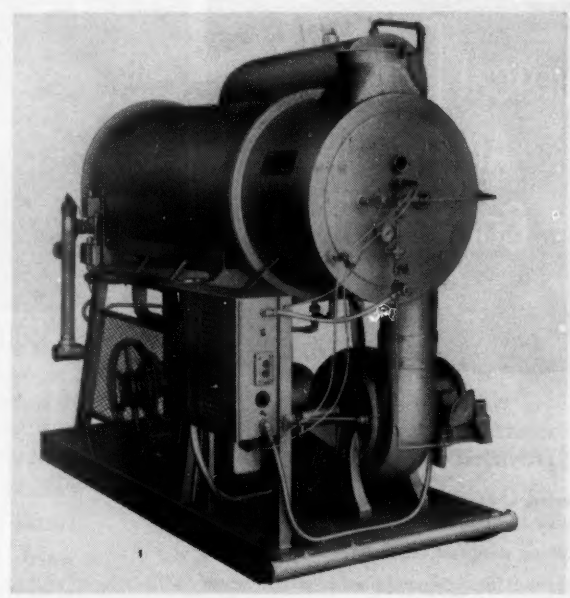
(*Continued from page* 18)

ates the pump speed to meet requirements of the system. The motor operates at a constant speed and the pressure control is connected to the building riser. Variation in demand governs the output speed of the drive and the capacity developed by the pump, thus maintaining a constant pressure. A complete package, the system consists of pump, motor, drive coupling and control.

Weil Pump Company, 1530 N. Fremont St., Chicago 22, Ill.

HOT OIL HEATERS

Three new models of Hi-R-Temp oil heaters have been added to this line, in 7,000,000, 13,000,000 and 300,000 Btu/hr input. By means of heat transfer oils, high temperatures to 600 F are achieved without the accompanying high pressures associated with steam plants. Units modulate automatically, require no water treatment, minimize problems due to freezing or corrosion and are unitized.



Vapor Heating Corporation, 80 E. Jackson Blvd., Chicago 4, Ill.

SPLIT-PHASE BLOWER MOTORS

New ½ and ¾-hp split-phase motors, specially designed for belt-driven blowers and fans, are now available in either 115 or 115/230-volt models. Units have a low starting torque and smooth acceleration, cited as improving performance and reducing maintenance, belt and bearing wear and starting noise. In addition, a new winding design for blower applications is cited as having eliminated the light flicker usually associated with higher hp split-phase motors.

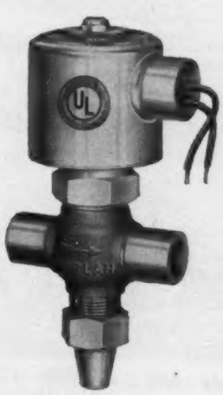
Westinghouse Electric Corporation, Industrial Motor Dept., P. O. Box 566, Lima, Ohio.

SPORLAN SOLENOID VALVES *are an*

OPEN and CLOSED case...

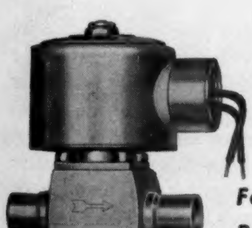
of outright Peak Performance — for all refrigeration and air conditioning applications

You be the judge... Sporlan solenoid valves are the best buy because of their high MOPD rating, the same valve type can be used on either Refrigerant 12 or 22 systems ... Extremely simple in design with few internal parts ... Sturdy construction ... Tight closing through use of synthetic seating material... Three sizes of high quality, power packed, layer wound, moisture-proof coils actuate the complete line.



When Sporlan Products are used exclusively, the evidence is conclusive.

There are Sporlan solenoid valves for the flow control of refrigerants, water and steam.



For complete presentation of the facts write for your copy of Bulletin 30-10 today.

SPORLAN VALVE COMPANY

7525 SUSSEX AVENUE ST. LOUIS 17, MISSOURI

EXPORT DEPT. 85 BROAD ST., NEW YORK 4, N. Y.

PARTS AND PRODUCTS (Continued)

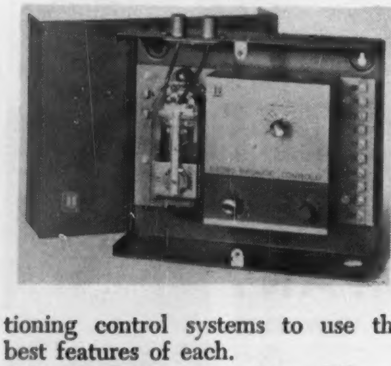
2-CU FT REFRIGERATOR

Rolling on casters, Model 2L-2-075 needs only connection to facilities to begin operation. Protected against

pressure build-up should the power fail, the unit re-starts automatically when facilities are restored. The cascade refrigeration system is cited as holding −130 F within ± 1 F. Other features include two fans for cooling the machine compartment and non-settling, rigid polyurethane foam pour-in-place insulation.
Harris Manufacturing Company, Inc., 308 River St., Cambridge 39, Mass.

TRANSDUCER

To convert temperature electronically into air pressure, the Mark III transducer has transistorized circuitry and etched circuit boards. With this unit, the sensitivity, flexibility and fast response of electronic thermostats can be combined with pneumatic operation, cited as permitting air condi-



tioning control systems to use the best features of each.

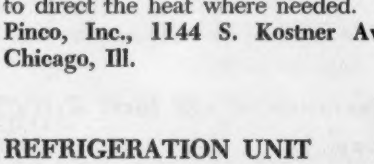
Inputs from up to three different thermostats can be taken by the transducer and outputs can be sent to pneumatic motors, dampers and valves. The unit also can link a pneumatic system to a centralized electronic control center to permit operation of an entire building from one remote location.
Minneapolis-Honeywell Regulator Company, 2727 S. Fourth Ave., Minneapolis 8, Minn.

SILENCERS

Featured on Conic-Flow silencers are a bell-mouth entrance for minimum entrance loss, a solid nose entrance for maximum noise reflection, narrow throat passage for maximum impedance and straight throat passage for minimum air friction. Diverging design of the evase exhaust is cited as keeping turbulence at a minimum and allowing for maximum pressure regain. Two types of silencer are available, for standard and low pressure.
Industrial Acoustics Company, Inc., 341 Jackson Ave., New York 54, N. Y.

INFRARED HEATER

By means of a fused quartz element, Quartz-Raye, Jr. is cited as producing heat in a short time under extremes of cold weather. For indoor or outdoor application, the 800-watt, 120-



volt unit is installed either to a control box already in a wall or by means of a plug-in cord, swivelling up or down to direct the heat where needed.
Pinco, Inc., 1144 S. Kostner Ave., Chicago, Ill.

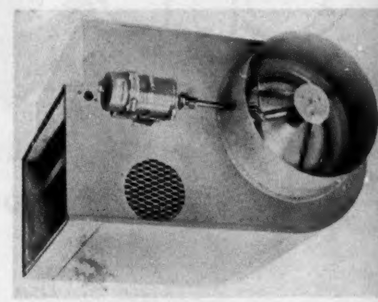
REFRIGERATION UNIT

Designed for use in delivery trucks, the Super Leader features a hermetic type, extended shaft, high capacity compressor linked to the power take-off by an automotive type, tubular drive shaft. This assembly mounts on a bracket on the truck chassis and supplies the refrigerant for the nose-mounted refrigeration package. The compressor and three-hp, 220-volt ac motor are on the same shaft.

In over-the-road operation, a magnetic clutch, controlled by a thermostat, automatically engages the compressor as cooling or winter heat is required. Self-sealing, flexible refrigerant lines connect the unit to the compressor.
Arctic Traveler, P. O. Box 989, Montgomery, Ala.

AIR MIXER

To provide multizone control for single-duct systems, provide a dual



effect for heat pumps and package air conditioning units using single-duct systems, rebuild static pressure to air-starved zones in dual or single duct systems and to reduce supply duct sizes on heating installations using high temperature air at low velocity are applications cited for the BMB Booster Air Mixing Unit. Applicability is to schools, medical buildings, stores and homes.
Barotherm Corporation, 1630 Industrial Park St., Covina, Calif.

CONTROL VALVE

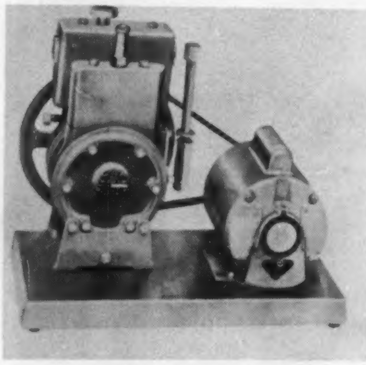
Designed to be used whenever low entering air temperatures will result in condensing head pressures too low for proper thermal expansion valve feeding and operation, the Unipressure Controller may be used on all condensing units and air-cooled condensers made by this manufacturer in the 2 through 50-ton capacity range. It may be utilized also for multi-circuit condensers with two or more compressors to achieve independent control of each circuit.

Fully automatic, this head pressure control valve requires no electric or pneumatic power, is explosion-proof and will function when mounted in any position. The unit is a self-contained, diaphragm-type valve which operates on the hot gas by-pass principle. Condensing capacity is modulated by flooding part of the condensing surface with liquid refrigerant. When head pressure falls, the valve permits some of the discharge gas

from the compressor to by-pass the condenser and enter the condenser outlet line and receiver. This restricts drainage of refrigerant from the condenser, flooding it to maintain the condenser and receiver pressure at the valve setting.
Trane Company, La Crosse, Wisc.

HIGH VACUUM PUMPS

Duo-Seal high vacuum pumps are offered in four sizes: 0.75, 2, 5 and 15-cfm capacities, each of which is



cited as drawing down to 0.1 micron efficiency. All models are vented-exhaust (gas ballast) types, which reduce pumpdown time and require few oil changes. A by-pass at the seal is provided to carry off the last increment of gas which may escape the exhaust port. This operation is repeated twice each revolution, or at least 600 times per min, reducing the pressure in the vessel connected to the intake, forming a greater vacuum each time.
Robinair Manufacturing Corporation, Montpelier, Ohio.

UNIT HEATERS

Available in ten basic sizes ranging in heating capacity from 25,000 to 250,000 Btu/hr input, NF gas-fired, pro-

peller-fan type unit heaters are recommended for automatically controlled applications in commercial or industrial installations. Compact and light in weight, the units require a minimum of space.

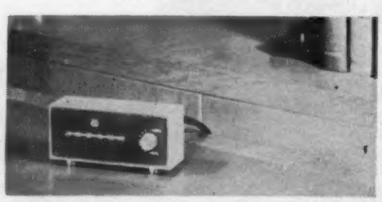
Electric drive motor for the fan is face-mounted and direct-connected and is cited as being trouble-free through elimination of the need for internal mechanical starting switches. Construction features of the units include die-formed, heavy-gauge furniture steel casing; aluminized steel heat exchanger tubes; raised-port cast iron burners, and a venturi mixing tube designed for optimum primary air injection, complete mixing of gas and air and minimum turbulence.
American-Standard, Industrial Div, Detroit 32, Mich.

FLOAT VALVE

Operating on a single pivot principle, this new valve has no moving parts. All rubber, the heavy-duty valve seat is adjustable and can be reversed. Also featured is a chrome-plated valve jet, cited as assuring positive shut-off. This valve has been incorporated in the manufacturer's line of float-type humidifiers.
Skuttle Manufacturing Company, Milford, Mich.

DESK THERMOSTAT

Pneumatically operated and readily adapted to most space temperature control operations, the Execustat is small in size, yet heavy enough to pre-

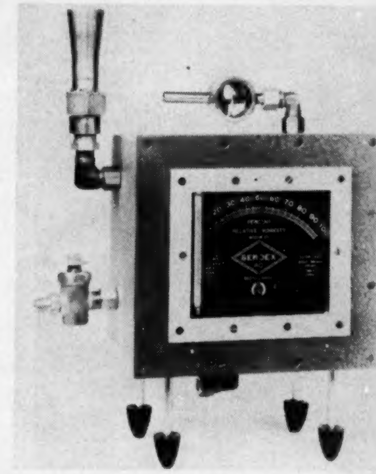


vent accidental brushing or tipping. Application is for heating and cooling, over a temperature range of 60 to 85 F. Response is ½ F and accuracy is within ± 1 F.
Powers Regulator Company, 3400 Oakton St., Skokie, Ill.

HUMIDITY INDICATOR

Principal use of this remote instrument is in instances requiring accurate exterior samplings of closed areas or containers. One outlet is connected to the area or container to be sampled; the other is connected to a vacuum line or pump. The reading may then be taken directly from a large dial, with no conversion charts or wiring involved.

Inside the air chamber of the gauge is a coil through which liquid of a predetermined temperature may be circulated to establish temperature equilibrium. Fluctuations caused by external conditions are retarded be-



cause of the double-wall, insulated construction of the unit.
Serdex, Inc., 91 Cambridge St., Boston 14, Mass.

POOL HEATER

Combination of automatic temperature control and balance of high velocity water flow through two tubes of the heat exchanger is cited as providing scale-free service. Additionally, this heater contains a built-in auxiliary by-pass that has a three-fold purpose: to improve filtration flow rates, assist the Unatherm Governor in providing automatic heater control under extreme conditions and enable the unit to be piped simply and economically. But two water connections, from the filter and to the pool, are necessary.

Six models ranging from 81,000 to 325,000 Btu/hr are now available with the Series II larger commercial models, ranging from 400,000 to 2,400,000 Btu/hr, already in use.
Raypak Company, Inc., 2231 N. Chico Ave., El Monte, Calif.

WALL-MOUNTED COOLING

For use either singly or in multiples for increased capacity, a new cooling unit for central air conditioning, the WispAir, supplies 18,000 Btu/hr cooling capacity. The blowers will deliver 600 cfm against a system resistance of 0.1 in. of water. Suited to houses, apartments, offices, stores and factories, the unit is designed for mounting outside on any wall. Factory-assembled, it includes a prewired thermostat and requires a minimum of ductwork.
Westinghouse Electric Corporation, Air Conditioning Div, Staunton, Va.

JOURNAL
OFFICIAL PUBLICATION

NOVEMBER 1961

Comment

OUR OWN "IT'S ABOUT TIME" DEPARTMENT

Morris Kaplan, whose Denver Symposium keynoting paper on Quality Control appears in this issue, as Technical Director of Consumers Union speaks from a wealth of opportunity to observe and to suggest product improvement in the interests of the purchaser. His observations were received attentively by his audience in Denver. We think he made some excellent points deserving of thought and action.

But we are far from convinced that merchandise that is less than satisfactory to some purchasers is a matter of considerable actionable concern to some of those responsible for it.

Several years ago it was observed authoritatively that "the big three" in the automobile industry expected 15% of their customers in any one model year to be to some degree unhappy with their automobile. It was stated that to raise the "contentment" factor to 90% would cost thus and so many millions of dollars to be added to the production costs of that year's model and thus affect incrementally the cost of each car produced. To raise contentment to 95% was considered hopelessly costly.

American productivity is allied inescapably with uniformity of output. Individual parts and components must come closer and closer to narrower tolerances in specifications but the assembly itself remains elusively unpredictable in certain ranges.

Be that as it may, it is only realistic that the manufacturer should plan for a probable life interval for his product and its specific components and for a level of acceptable quality performance as related to the expense of producing it.

There is nothing new about any of this. Perfection we cannot achieve, much less afford. The question becomes how much quality can we afford to provide and still remain competitive and solvent.

Yet there is no topic which manufacturers are much more reluctant to discuss than the specifically planned life expectancy for their individual products or the "contentment" levels that they can afford to meet; despite the fact that these are inescapable factors in their administrative procedures.

We do not know that their world is ready for such frankness but it might clarify matters to give more open recognition to the need for it.

Edward R Searles
Editor

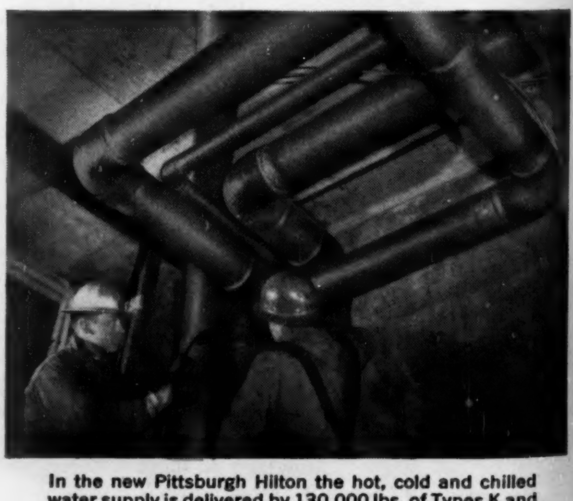
PITTSBURGH HILTON HOTEL, Pittsburgh, Pennsylvania. Architect: Wm. B. Tabler, N.Y.C.; General Contractor: Turner Construction Company, N.Y.C.; Mechanical Engineers: Jaros-Baum-Bolles, N.Y.C.; Plumbing Contractor: Sauer, Inc., Pittsburgh.

130,000 lbs. of water lines!
Anaconda copper tube up to 8″ throughout

There are many reasons why Anaconda Copper Tube is so often the choice for modern water supply and drainage systems. It is, for example, extremely easy to handle in limited work spaces. This, of course, helps keep initial costs low. Add such advantages as long life and trouble-free maintenance and you have a material mighty hard to beat.

For complete information about Anaconda Copper Tube and Fittings for general plumbing, heating, air conditioning and refrigeration, write for free copy of Publication B-1, Anaconda American Brass Company, Waterbury 20, Conn. In Canada: Anaconda American Brass Ltd., New Toronto, Ont. 61-1027

ANACONDA®
AMERICAN BRASS COMPANY



In the new Pittsburgh Hilton the hot, cold and chilled water supply is delivered by 130,000 lbs. of Types K and L Anaconda Copper Tube in sizes to 8″.

QUALITY is what the consumer wants

MORRIS KAPLAN

It is my observation that the term "quality" often is used loosely and in many different ways. Sometimes, it is a synonym for "high-priced"; sometimes it implies "durable," or "reliable"; sometimes "exclusive." But when the real sense in which the word is used is related to the specific user, I have found that it always means "desirable," however that user defines desirable for himself. The quality characteristics of a product, thus, seem to be those which a consumer wants in that product.

But it is also a fact that what a consumer wants in a product is often a reflection of his sophistication – his previous experience with it, his understanding of the needs it satisfies, and the degree of effectiveness with which these needs are fulfilled. Thus, if you try to find out what influenced a consumer to buy a specific refrigerator (as one measure of what he wants in the appliance), such factors as these are likely to crop up often: size, cost, presence of special features (magnetic doors, adjustable door shelves, unusual ice-cube maker, swing-out shelves, etc.), style and color, his opinion of the manufacturer's reputation, and automatic defrost feature.

There is likely to be no mention of how low and how uniform are the temperatures in the fresh food space, freezing compartment, and door-shelves; what happens to food temperature during the defrost cycle; the true, useful capacity of the refrigerator; its operating cost; its noisiness; ease of cleaning, etc.

Is this because the consumer is not interested in knowing that it costs twice as much to operate one refrigerator as another of comparable size and type (enough, incidentally, to go a long way toward paying for a second refrigerator during its life) or that the cost of operation of the convenient feature which permits the operation of the refrigerator-freezer without frost accumulation is half again as much as the cost of running a comparable manual defrost model? No. It is just that he doesn't know enough to ask such questions. And if he did, he wouldn't be able to get any answers.

But if he knew what questions to ask, his definitions of quality – of what he considers desirable – might be quite different from the idea he has developed out of his necessarily limited experience and his limited access to the experience of others.

> . . . The consumer just does not know enough to ask significant questions; if he did, he would not be able to get the answers.
>
> . . . desirable, or made to believe to be desirable?
>
> . . . You can sell quality, if the consumer can recognize it.
>
> . . . Nothing will convince a consumer that a manufacturer means what he says, better than a foolproof guarantee.
>
> . . . demise for the annual model change.

AS A PRACTICAL MATTER

How, then, should a manufacturer look at quality? If he were to ask his engineers to design in accordance with their concept of a high-quality refrigerator, with no restrictions imposed by sales managers, he would get a most unusual product. For his engineers, we may assume, are broadly knowledgeable, and therefore would place emphasis on characteristics which derive from the engineer's understanding.

Conversely, if he were to request that his market researchers determine what the prospective purchaser wants in a refrigerator, the list would be notably different from that prepared by the engineer. In fact, the prospective purchaser's list is likely to reflect, largely, the most recent advertising claims for refrigerators – what he has been conditioned to believe is desirable. His list represents the existing demand; the engineer's represents what the demand might be if consumers knew more about what the appliance does, and what it could do.

With these few comments on the meaning of quality as a background, let us now look at appliance quality in the real world, and consider the important aspects of Design, Production and Service.

Generally, a manufacturer chooses to incorporate into his design those characteristics which the consumer has already accepted as important, as well as those new features which he thinks he can easily convince the consumer to buy. Others, which are in fact more important, often go by the board because they involve teaching their importance to the consumer. And this is hard to do in a flashy TV commercial, or a splashy full-page ad. Furthermore, such claims are often hard to prove, and people have grown most skeptical.

It is many years since anyone has tried to sell a refrigerator with

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Morris Kaplan, Technical Director, Consumers Union, was the keynoter at the Domestic Refrigerator Engineering Symposium where the topic was "Organizing for Quality" at the ASHRAE 68th Annual Meeting in Denver, last June.

"half as costly to run" as its competitive claim. Who would believe the mere statement of the claim, and how could it be demonstrated convincingly? Similarly, a durable door liner or shelf or freezer-door or thermostat appear not to be competitive with the "square look" as a selling point. "Easy to clean" may be demonstrated effectively perhaps, but how does it compare in glamour to the "latest colors"? No one tries to sell "good regulation" or "uniformly low temperatures throughout the fresh food space, including door-shelves" or "frozen foods maintained between −1 and +1 F." It is easier to sell "no-frost" features, or "swing-out shelves" or "continuous ice-cubes."

I make no value judgments on the relative merits of these alternatives. Some of the advertised features may be truly more desirable than the ones not promoted. **My point is that the consumer never really has a chance to decide on the relative merits of the alternatives.** I am convinced that if consumers could, at the same price, have a choice between two refrigerators—one with a zero F freezer but with a relatively high cost of operation, and another with a 5 F freezer but with a correspondingly lower cost of operation, there would be customers for each. But generally he does not have such a choice. The manufacturer makes the decision for him. And since sales are paramount for the manufacturer, the true consumer interest is not always served optimally. The manufacturer may, for example, provide a 10 F freezer in a new-look, costly-to-manufacture cabinet.

THE BASIC PROBLEM

Is the manufacturer making what the consumer would want if the consumer knew what the technological alternatives really were? — that puts the problem in the broadest and most fundamental terms. When people say, as they have been saying increasingly in recent years, "they don't make them like they used to" they are usually not referring to this problem. They are talking about the quality of the product the manufacturer does produce. To a group such as yourselves, I need hardly labor the validity of this complaint. The trade press and, I am sure, the management of the appliance companies have discussed it at length. But until the condition is corrected, it cannot be stated too often. Let me therefore do my part by quoting our own experiences at Consumers Union and a few of those communicated to us by our readers.

In our recent study of the "no-frost" refrigerator-freezer combinations, **7 out of 16 models tested were defective as received, or developed defects under normal use during the tests!** Four of these were unable to maintain satisfactory temperatures because the fan did not circulate air through the unit. (In one case, the fan blade was loose on the shaft; in the three others, there were obstructions of one kind or another which prevented proper circulation of air.) One model had a defective thermostat; another a faulty defrost mechanism; another had improperly insulated refrigerant tubing, resulting in drippage of condenser water on the floor.

And our mail is full of many such specific performance comments and complaints.

Perhaps you would derive some comfort from the fact that this problem is not limited to refrigerating appliances. Of 36 models of washing machines tested, 14 were defective as received or developed early defects in normal use. Three models leaked, four gave timer trouble, one did not fill properly, another did not empty properly, one did not agitate properly, one was especially noisy — all in all, a shameful record.

So, having bought a product which was not designed to meet his needs optimally, and finding it defective either on receipt or shortly thereafter, the consumer hopefully calls for service. I have read hundreds of bitter, frustrated, angry letters on this subject.

Our direct experiences with appliance servicing have been in the same vein. Although our engineers are often in a position to repair an appliance far more rapidly and effectively than a serviceman, our normal procedure is to call an authorized service dealer. One common reaction is "there's nothing wrong; they all behave that way" — even when the appliance is obviously not functioning in accordance with the design intent. Another is to attempt to repair the appliance, with an apparent lack of familiarity with its components and circuitry. "I've never seen one of these before," says the serviceman, with evident interest, but to our dismay. So he fiddles and fools, doing his best to improvise. On one occasion, such improvisation resulted in an ingenious modification and improvement of the original design. But the more usual outcome is a less happy one — either the service manager or the factory representative is called in by the dealer, or we buy another sample when the defective one can not be repaired effectively.

Even when the serviceman can spot the trouble, if it turns out to be a defective part requiring replacement, it sometimes takes weeks before it can be obtained. And our correspondence indicates that this situation is true not only for new models, but for old ones as well.

Now, the few examples I have cited and the many more I could describe are not entirely the fault of the servicemen. The manufacturer who is concerned with quality in its broadest sense will also accept his share of responsibility for handling the service problem. He will design so that service problems are reduced. He will make servicing easier. He will arrange for rapid availability from a complete inventory of parts on all models. And if these requirements seem too difficult and too expensive, he will give some thought to alternative solutions to achieve the same end.

WHERE TO LOOK FOR SOLUTIONS

I think the manufacturer desirous of making a quality product ought first to define quality in terms of function, not frill. Let him first define optimum performance, convenience, economy, maintenance and safety in terms of what he or his engineers and home economists would want for themselves, and then let him add on style, gadgetry, frills, extras. "But," I can hear someone saying, "this will raise the cost, and consumers buy by price; not quality."

I don't believe it. I think you can sell true quality if the con-

(*Continued on page* 81)

Air Infiltration through revolving doors

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Air infiltration through doors may cause discomfort in areas near the doors, and thus it is desirable to know the rate of infiltration for proper sizing of the heating and cooling equipment. A recent research (1) supplied comprehensive information on air infiltration through swinging doors, but only limited data (2) are available on air infiltration rates through revolving doors. The purpose of the present investigation is to provide additional information on this subject. The research has been carried out at the ASHRAE Research Laboratory under the guidance of the former Research Advisory Committee on Heating and Air-Conditioning Loads. Experiments were made both under heating and cooling conditions. With the motor-driven revolving door, the door speed and temperature and pressure differences between indoors and outdoors were recognized as significant factors influencing the air-infiltration rate. With manually operated doors, however, additional information regarding a representative traffic pattern and the corresponding average door rpm was needed. The amount of air turbulence inside and outside, and the condition of the door seals would influence infiltration to a certain extent. The research was planned to determine the influence of all these variables on air infiltration.

TEST SETUP

A plan view of the test setup is shown in Fig. 1. A revolving door, 6 ft - 4 in. diam and 6 ft - 10 in. high was installed just inside a double door entrance to the Laboratory. When the swinging doors were opened, a typical revolving door entrance was available for testing. The door was motor driven and its speed could be adjusted by means of a variable-pitch pulley. The door could also be operated manually.

A plywood partition was erected across the entrance back of the door, and all cracks in the test room were caulked to make the enclosure as nearly air tight as possible.

The exhaust system E served several functions. It provided a means of producing any desired pressure differential across the revolving door, it exhausted the infiltration air which, in a normal

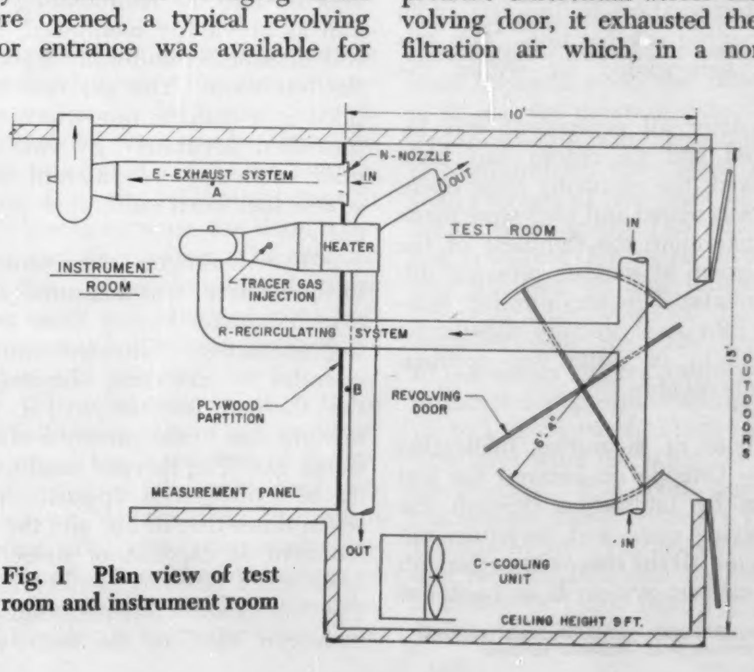


Fig. 1 Plan view of test room and instrument room

L. F. Schutrum was Research Supervisor, C. M. Humphreys was Assistant Director of Research and J. T. Baker was a Research Engineer at the ASHRAE Research Laboratory. N. Ozisik is with the Oak Ridge National Laboratory. This paper was presented at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

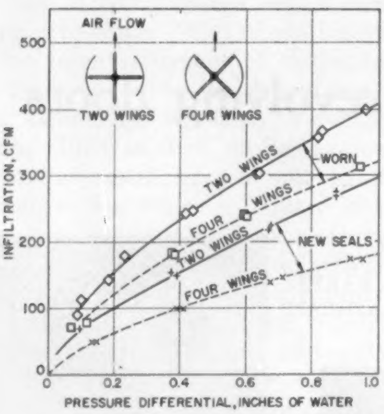


Fig. 2 Infiltration through new and worn* door seals (door not revolving)

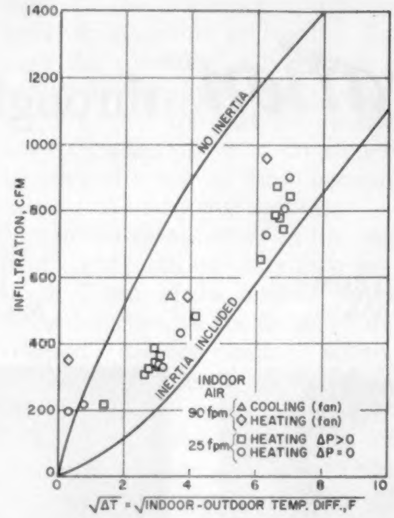


Fig. 3 Observed data and calculated curves of infiltration through revolving door at 10 rpm (air leakages deducted)

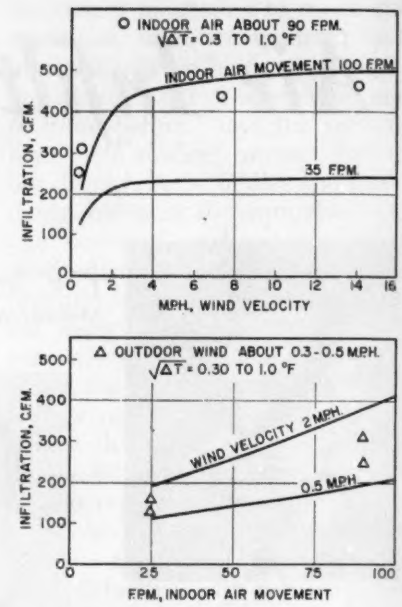


Fig. 4 Effect of wind and indoor air movement on infiltration for small temperature differences and door speed of 5 rpm (air leakages deducted)

installation, would diffuse to other parts of the building, and a nozzle at the duct entrance provided a means of measuring the quantity of the exhausted air.

The recirculating system R also served several purposes. The air drawn from the test room was discharged through a heating coil, thus providing a means of controlling the temperature in the test room during the heating season. The tracer gas was injected into this system between the fan discharge and the heating coil. The fan handled sufficient air, and the supply and return ducts were so located, that the tracer gas was well distributed and the air temperature in the space was reasonably uniform.

A cooling system also was provided to permit a reduction of the air temperature in the test room during the summer tests. This system recirculated air within the space.

After all equipment was installed and all cracks had been caulked, the revolving door opening was sealed and tests were made to determine the tightness of the test room at various pressure differentials. The uncontrolled leakage rate at a pressure differential of 0.5 in. of water was approximately 40 cfm.

Methods of measuring infiltration air — Outside air entered the test room by infiltration through the revolving door, and an equivalent amount left the space either through the exhaust system E or back out through the revolving door. These two routes are shown diagrammatically in Fig. 1-A of Appendix A. The problem was to devise measurement methods by which both of these flows could be determined. It was decided that this could best be accomplished by measuring the total infiltration by the tracer gas technique, and measuring the flow through the exhaust duct by means of a nozzle. The flow back through the door could then be determined by difference.

In applying the tracer gas method of measurement, the tracer gas, hydrogen in this case, was injected into the recirculating system as previously mentioned, and was distributed uniformly through the test room. The gas was supplied at a uniform rate which was measured accurately by two all-glass rotameters of different sizes which had been calibrated previously.

The hydrogen concentration in the room air was measured with a katharometer having three sensing elements. This instrument operates by detecting changes in the thermal conductivity of the mixture due to the presence of the tracer gas. The thermal conductivity of hydrogen is approximately seven times that of air, and the instrument is capable of detecting quite small amounts of the tracer. The maximum concentration of hydrogen used in the tests was approximately 1%, which is well below the lower limit of flammability.

The scale of the katharometer was graduated to read in per cent of helium. It was therefore necessary to recalibrate the instrument for hydrogen. This was done by mixing known volumes of air and hydrogen in a large container, and recirculating the mixture through the sensing elements of the instrument.

A preliminary survey indicated that the concentration of hydrogen was quite uniform throughout the test room except in an area within about 3 ft of the revolving door opening, where the mixing of the room air and infiltration air was taking place. In all later tests, the hydrogen concentration was measured at the 72-in. level at point A in the test room and at point B in the exhaust duct. In tests with the exhaust fan operating, the concentration at both points was essentially the same (see Fig. 1).

Rapid diffusion of hydrogen into the air in the inside segment of the door would introduce errors in the results. However, estimates indicated that this diffusion rate was quite small. The method of calculating the infiltration rate from the tracer gas concentrations is explained in Appendix A.

* Note: Although worn, the seals provided good contact with adjacent surfaces.

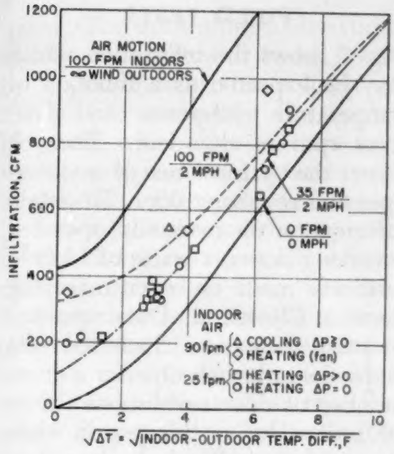


Fig. 5 Infiltration through revolving door at 10 rpm (air leakage through seals deducted)

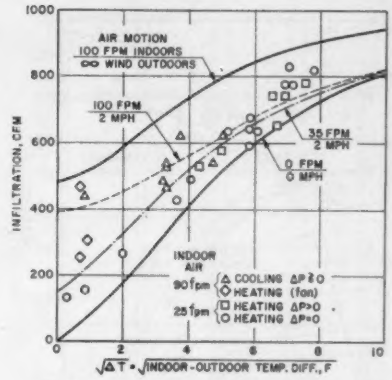


Fig. 6 Infiltration through revolving door at 5 rpm (air leakage through seals deducted)

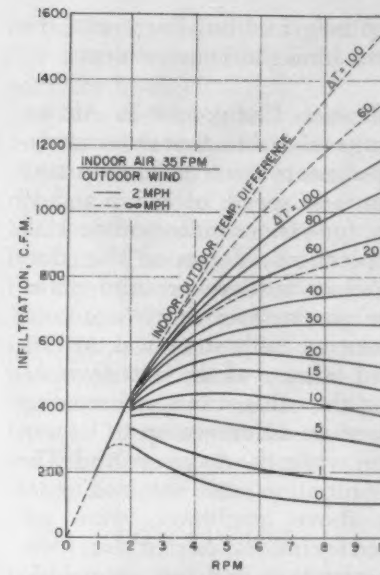


Fig. 7 Infiltration vs. rpm and indoor-outdoor air temperature difference (air leakage past seals deducted)

Flow in the exhaust system was determined by a nozzle located at the duct inlet as shown in Fig. 1. The nozzle size was varied from 1 to 7 in., depending upon the rate of flow to be measured.

Other instrumentation—Differential pressures were measured with draft gauges. The outdoor pressure was obtained with a small static tube extending approximately 3 in. from the face of the wall.

The air temperature at several locations and elevations in the test room was sensed by copper-constantan thermocouples. These were connected to an indicating electronic type potentiometer.

A vane type anemometer located about 5 ft outside the door opening was used to measure the wind velocity. Air velocities in the test room were measured with a heated-thermocouple type anemometer about 3 ft away from the door.

TEST PROCEDURE

After all equipment was in operation and final adjustments were made for door speed, room air temperature and pressure, enough time was allowed for the room air temperature to reach steady conditions. This usually required from one to two hr. Hydrogen was then introduced into the room at a rate which usually would give a reading on the upper half of the katharometer scale, which for hydrogen covered a range from 0 to 0.7%. The actual test would be started, about 15 min later, after the hydrogen concentration level had become stable.

During the 15-min test period, air temperatures and pressures, and rotameter and katharometer readings were taken at fixed intervals. The wind velocity and the door speed also were recorded.

Tests were made under both heating and cooling conditions. Indoor-outdoor air temperature differences up to 60 F could be obtained in winter, while in summer the indoor temperature could be held as much as 30 F below the outside air.

The test program was planned to permit the evaluation of all of the important variables, including door speed, indoor-outdoor temperature and pressure differentials, indoor air motion and outdoor wind.

Analysis of air infiltration through revolving doors – For the purpose of analysis, air infiltrating through a revolving door can be separated into two components: One, component A, is the infiltration through the cracks between the door housing and door wings. The amount of this component depends upon the width and length of crack and the pressure difference between indoors and outdoors.

The other component, B, is the air infiltration related to door movement. When a segment of the revolving door filled with cold outdoor air is turned to the warm room side, circulation starts between the room and the segment air due to the density difference. A similar air exchange takes place at the opposite segment of the door with the outdoor air. The amount of this component of infiltration depends upon indoor-outdoor temperature difference, door speed and door size. Furthermore, indoor air motion and outdoor wind were expected to influence this infiltration. An analysis of this problem and the derivation of an analytical equation for calculating air infiltration related to the door movement is included within Appendix B.

TEST RESULTS

Infiltration Component A: Air leakage past door seals due to pressure differential – Infiltration past door seals depends upon the closeness of fit and the pressure differential. To investigate the magnitude of this infiltration, tests were made with two sets of new seals and one set of worn seals. The door was not revolving for these tests but was adjusted so that either two wings or four wings of the door were touching the door housing. Fig. 2 shows the magnitude of infiltration for pressure differentials up to 1 in. of water. These infiltration rates were corrected for room air leakage. With both new and used seals there were no visible cracks. Where visible, crack size can be estimated and the infiltration may be found from Fig. 10 of Reference 1. Data on pressure dif-

ferentials for buildings can be found from the same reference.

Infiltration Component B: Air exchange related to door movement— Experiments were made with constant door speeds of 1, 2, 5 and 10 rpm, for various indoor-outdoor air temperature differences. The effect of indoor-outdoor pressure difference on the door movement component of infiltration was investigated in some of the tests by operating the exhaust fan and creating a pressure difference up to 1 in. of water while the door revolved. The air infiltration rates, obtained under the above conditions, were adjusted for infiltration past seals (two-wing contact) and test room leakage due to pressure difference, and the results were plotted in Fig. 3 as a function of indoor-outdoor air temperature difference for a constant door speed of 10 rpm.

These plotted values were found to be practically independent of the pressure differential. This is shown in Fig. 3 by the proximity of the squares and circles which are the symbols for tests with and without pressure difference. The upper curve of Fig. 3 shows calculated infiltration rates neglecting the inertia of the air, and assuming still air conditions both indoors and outdoors. In calculating the lower curve the effect of inertia was included. This curve follows the observed data but lies below the points.

However, in the experiments, air motion indoors and wind outdoors were expected to increase the infiltration. Moreover, summer data showed higher infiltration rates than the winter data. The only difference in the test setup was the cooling blower used in the summer tests which caused higher indoor air velocities. The blower was then used for heating tests (coil not cooled) and the effect of air velocity on infiltration was verified. The experimental points in Fig. 3, representing the data with indoor air movement averaging about 90 fpm with the door stationary (triangles for summer and diamonds for winter data), are higher than those in which the average velocities were about 25 fpm (squares and circles).

Wind velocities 5 ft outside the door were quite low, usually less than 2 mph, while at about 10 ft from the door limited data show the velocity to be about twice as high. The data in Fig. 4, for quite small indoor-outdoor temperature differences, show the effect of wind velocity and indoor air movement on infiltration rates. The curves were calculated as described in Appendix B, based on the assumption that the heads due to the average non-directional air movement and wind velocities have the same effect on infiltration as equal heads due to temperature differences. The trends, if not precisely the magnitude, of the calculated curves in Fig. 4 are in agreement, and thus this premise was assumed to approximately represent the effect of wind and air motion.

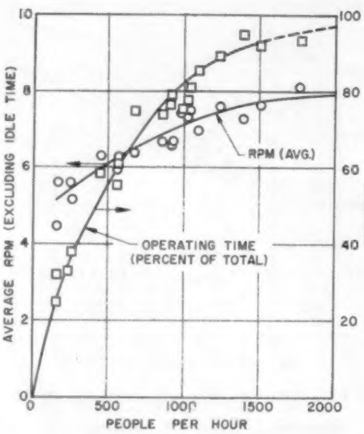


Fig. 8 Operating time and average rpm vs. traffic rate for seven manually operated revolving doors

Figs. 5 and 6 show experimental points, and curves calculated for various indoor air movement and outdoor wind velocities. The approximate indoor air movement for the test points is indicated by the symbols; however, the wind velocities are not identified.

The infiltration rates shown by the solid curves in Fig. 7 are for an indoor air movement of about 35 fpm and a wind velocity of 2 mph. The maximum wind velocity normal to the entrances at the curb line was reported by T. C. Min[1] to be about 2 mph when the wind velocities at the U. S. Weather Bureau were 20 mph. The dashed curves represent the infiltration with extremely high wind velocities which might cause one hundred per cent air exchange in the outside segment of the door.

FIELD TESTS

Fig. 7 shows the infiltration which may be expected as a function of temperature difference and constant rpm of the door. Thus, it shows the performance of a motor-operated revolving door. To obtain information on manually operated revolving doors, a series of 19 field tests was made on seven revolving doors in Cleveland. Data recorded by two observers included the elapsed stop watch time for a given number of door revolutions (50 or 100), time during this period when the door was idle, and a count of the number of people passing through in each direction. The traffic rates varied from 160 to 1770 people per hr. The two curves of regression in Fig. 8 show the relationship found from the field tests between traffic rate, and average rpm and operating time.

Infiltration (Component B) through manually-operated and motor-driven revolving doors — The infiltration through manually operated doors for a given traffic rate was taken as the product of the operating time from Fig. 8 and the infiltration from Fig. 7 determined for the average rpm given in Fig. 8. To substantiate this method, three tests were conducted at the Laboratory with summer cooling conditions and with Laboratory personnel passing through the revolving door, manually operated, at traffic rates ranging from 1075 to 1975 people per hr. While this random traffic pattern was maintained, the tracer gas was being used to measure the infiltration. The observed infiltration, 400-450 cfm, was lower by about 10% on the average than that predicted.

The infiltration through manually operated revolving doors is shown in Fig. 9. The wind velocity was taken as 2 mph, but a few dashed curves for maximum wind velocity are given to show the probable upper limit of infiltration.

For motor-driven doors the infiltration rates can be found from Fig. 7.

EFFECT OF PEOPLE

In a series of tests the doors were motor operated with none, one, two or four of the door segments occupied by people walking around continuously without leaving the

space. The infiltration under these conditions was found to be lower by approximately the volume of the people occupying the space. However, when a person steps into the door space, he displaces part of the air in the space, and when he leaves the space on the opposite side of the door, an equal volume of air must replace him. This effectively increases the infiltration, counteracting the decrease caused by the occupancy of the door space. Calculations show that the effect of people on infiltration is small and can be neglected.

Example of infiltration calculation—

Given: A building has a pressure differential of 0.46 in. of water when the indoor air is maintained at 75 F and the outdoor at zero F. Find: The infiltration through a manual revolving door when the traffic rate is 1000 people per hr.

Solution: From Fig. 9 the infiltration rate due to the temperature difference of 75 F is 750 cfm. The infiltration through door seals due to the pressure differential of 0.46 in. of water is 250 cfm from Fig. 2 for two wings and worn seals. The total infiltration is 750 + 250 or 1000 cfm.

Second Example—

Given: The same conditions as in the previous example except that the door is motor driven and operates at a constant speed of 10 rpm.

Solution: The infiltration due to a temperature difference of 75 F is 1040 cfm as taken from Fig. 7. The infiltration due to a pressure difference of 0.46 in. of water is the same as in the first example, making the total 1040 + 250 or 1290 cfm.

DISCUSSION

All calculated infiltration curves and experimental data were based on the standard air density of 0.075 lb per cu ft. The calculated curves were not adjusted for the change in volume of the infiltration air due to temperature change as it passes from the outdoors to the indoors. Moreover, the infiltration rates found in the various figures which are a function of temperature difference were based on winter conditions. The same temperature differences for summer conditions would correspond to slightly lower density differences. It was felt that the refinements for these changes in air density were not necessary and would only complicate the presentation of the results.

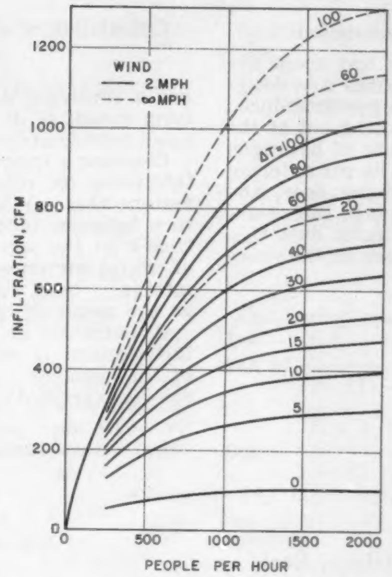


Fig. 9 Infiltration through manually operated revolving doors (air movement 35 fpm indoors, air leakage past door seals deducted)

One would expect that the motion of the door would tend to increase the crack area between door seals and the adjacent surfaces and thus increase the air leakage past the door seals due to pressure difference. The data showed only insignificant changes in crack leakage with door movement.

The effect of air movement indoors and wind outdoors on infiltration was not rigorously investigated, as this would likely require an extensive research project in itself. However, the assumption that air velocity heads in the boundary spaces were equivalent in effect to heads produced by temperature difference is a correction in the proper direction to the infiltration calculated for still-air conditions.

The centrifugal force of the air in the door segment was estimated from the difference in density of the incoming and outgoing air of the segment and the door speed. This effect was neglected in the final analysis since the magnitude of the force even at the higher rpm (10) was less than 10% of the temperature head.

In a few tests, heated air was introduced into the inside segment of the revolving door, through the opening provided for a luminaire in the ceiling of the revolving door housing. The rate of air entering through this opening was about 250 cfm. With but extremely limited data, a definite conclusion cannot be drawn; however, the infiltration rates deduced from these tests at 10 rpm were close to the values given by the upper curve of Fig. 5.

CONCLUSIONS

Infiltration through a revolving door can be estimated by combining the air leakage past the door seals with the infiltration related to the revolving of the door. The magnitude of air leakage through the seals of the door is the result of the pressure differential across the building entrance and the size of the openings at the seal. Revolving the door causes an exchange of indoor and outdoor air of approximately equal volume. The amount of this air exchange depends upon the door speed and the temperature differential and somewhat upon the wind and indoor air velocities.

Infiltration due to air leakage past the seals of the door is given in the paper as a function of the indoor-outdoor pressure differential. Infiltration related to the door movement also is given for a motor-driven revolving door, and for a manually operated door for traffic rates up to 2000 people per hr.

Data given in this paper are based on the use of door seals which provide good contact with the adjacent surfaces. If seals deteriorate to the point that good contact is not maintained, leakage past the seals will increase greatly.

ACKNOWLEDGMENT

The authors recognize the valuable guidance of Messrs. A. M. Simpson and R. G. Chapman in setting up the research program, and also wish to thank the Revolving Door Div of the International Steel Door Company for its cooperation in supplying the revolving door for these experiments. Helpful suggestions from the staff of the Laboratory are acknowledged gratefully.

APPENDIX A

Infiltration rates from tracer gas measurement

The hydrogen and air flow-circuits in the test room are shown in Fig. 1-A. The hydrogen flow paths are designated as Q_{HT} the total steady rate of hydrogen introduced into the room, Q_{HD} the rate of hydrogen flowing out of the room through the duct, and Q_{HO} the net rate of hydrogen passing out through the revolving door. The air entering the test chamber is Q_{AI} through the revolving door, and Q_L the leakage through walls, etc. Air leaves the chamber via the revolving door Q_{AO} and the exhaust duct Q_{AD}. C_s and C_D are the concentrations of hydrogen in the space and in the exhaust duct.

By volume, balances on

hydrogen $Q_{HT} = Q_{HO} + Q_{HD}$ A-1

and on air $Q_{AI} = Q_{AO} + Q_{AD} - Q_L$ A-2

The measured concentrations are

$$\text{in space } C_s = \frac{Q_{HO}}{Q_{AO} + Q_{HO}} \qquad \text{A-3}$$

where from Eq. A-3 and A-1

$$Q_{AO} = \left(\frac{1 - C_s}{C_s}\right) Q_{HO} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - Q_{HD}) \qquad \text{A-4}$$

and in duct

$$C_D = \frac{Q_{HD}}{Q_{AD} + Q_{HD}} = \frac{Q_{HD}}{Q_D} \qquad \text{A-5}$$

from which

$$Q_{HD} = C_D Q_D \qquad \text{A-6}$$

from A-5 and A-6

$$Q_{AD} = (1 - C_D) \frac{Q_{HD}}{C_D} = (1 - C_D) Q_D \qquad \text{A-7}$$

Substituting Eq. A-6 into A-4

$$Q_{AO} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - C_D Q_D) \qquad \text{A-8}$$

and substituting Eq. A-7 and A-8 into A-2

$$Q_{AI} = \left(\frac{1 - C_s}{C_s}\right) (Q_{HT} - C_D Q_D) + Q_D (1 - C_D) - Q_L \qquad \text{A-9}$$

The infiltration was determined from Eq. A-9 for observed values of C_s and C_D from katharometer measurements, Q_{HT} from rotometer readings, and Q_D from nozzle information. Q_L was determined by measurement and its relation to indoor-outdoor pressure differences. For tests without pressure differences the exhaust volume Q_D was zero and Q_L was assumed to be zero.

APPENDIX B

Calculation of air infiltration component B through a revolving door

Let a revolving door having 4 segments rotate at a uniform speed, N, in the direction shown in Fig. 1-B. Assume outside air is colder than room air.

Consider a quarter-section OAB having its leading edge OA along ox reference axis $\theta = 0$. At this moment, assume that air in the segment has a uniform temperature between indoor and outdoor air temperatures. At time θ let the segment be at position OA_1, and the volume of warm air entering into the segment during the time interval θ be q. Assume air displacement is taking place as the result of buoyancy due to cold air in the segment and warm air in the room, and that warm air entering the segment is collecting in a uniform layer at the top of the segment. At the time θ, the rate of change of q can be expressed as follows:

$$\frac{dq}{d\theta} = CA\sqrt{2gh} \qquad \text{B-1}$$

where

A = area of opening

h = head of air

C = flow coefficient

In this equation, both A and h are functions of θ and q. Therefore, in order to solve equation B-1, A and h are to be expressed in terms of q and θ. As area, A, first increases and then decreases during a complete cycle, the equation can be solved separately for the two time intervals defined as follows:

1. "Opening-cycle" corresponding to the time interval:

$$\theta = 0 \text{ to } \theta = \frac{1}{4N}$$

2. "Closing-cycle" corresponding to the time interval:

$$\theta = \frac{1}{4N} \text{ to } \theta = \frac{1}{2N}$$

First, consider the opening cycle. For this cycle, h and A can be related to q and θ as follows:

$$h = h_o \frac{V - q}{V} \qquad \text{B-2}$$

$$A = \tfrac{1}{2} L (2\pi RN\theta) \frac{V - q}{V} \qquad \text{B-3}$$

where

h_o = head when $\theta = 0$

V = Volume of the door segment

Substituting B-2 and B-3 in B-1:

$$\frac{dq}{d\theta} = CL\pi RN\theta \frac{V - q}{V} \sqrt{2gh_o \frac{V - q}{V}} \qquad \text{B-4}$$

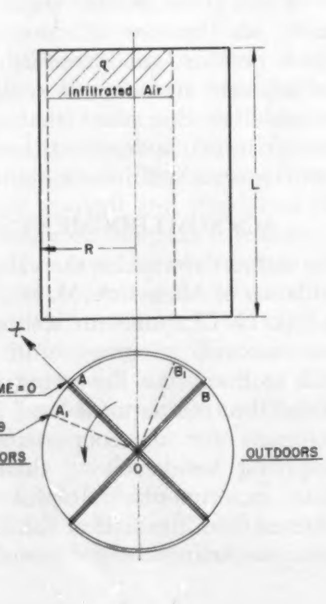


Fig. 2B Air exchange from one side of revolving door into door segment (includes inertia effect)

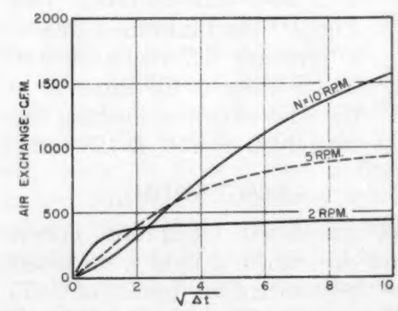


Fig. 3B

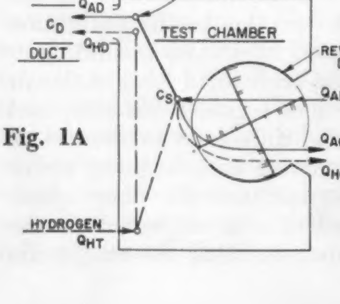
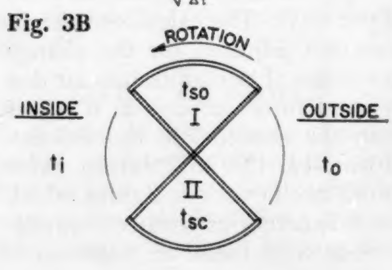


Fig. 1A



During the closing-cycle the time interval is $\frac{1}{4N} < \theta \leqq \frac{1}{2N}$.

For convenience take $\frac{1}{4N}$ as zero θ for reference purpose, thus the value of A for this closing cycle is

$$A = \tfrac{1}{2} L\, 2\pi RN \left(\frac{1}{4N} - \theta\right)\frac{V-q}{V} \qquad \text{B-5}$$

and equation B-4 becomes

$$\frac{dq}{d\theta} = CL\pi RN\left(\frac{1}{4N} - \theta\right)\frac{V-q}{V}\sqrt{2\,gh_o\frac{V-q}{V}} \qquad \text{B-6}$$

Inertia effect—In the foregoing derivation the inertia of air in the segment, that is, the head lost in accelerating this mass of air from initially zero to some finite velocity, was not considered. In a door segment, the inertia head at a time θ can be approximated as follows:

$$h_i = \frac{1}{2}\,\frac{L}{g}\,\frac{dv}{d\theta} \qquad \text{B-7}$$

where L/2 is used to be consistent with h_o, which is described later

since

$$vA_s = \frac{dq}{d\theta} \text{ or } \frac{dv}{d\theta} = \frac{1}{A_s}\,\frac{d^2q}{d\theta^2}$$

and

$$A_s = \frac{V}{L}$$

then

$$h_i = \frac{L^2}{2\,g\,V}\,\frac{d^2q}{d\theta^2} \qquad \text{B-8}$$

and the actual head causing infiltration at any time θ would be the buoyancy head given by equation B-2 less the inertia head given by equation B-8, that is

$$\text{Actual head} = h - h_i = h_o\frac{V-q}{V} - \frac{L^2}{2\,g\,V}\,\frac{d^2q}{d\theta^2} \qquad \text{B-9}$$

Equation E-4 for the opening cycle becomes:

$$\frac{dq}{d\theta} = K\theta\frac{V-q}{V}\sqrt{\frac{V-q}{V} - \frac{L^2}{2\,gV\,h_o}\,\frac{d^2q}{d\theta^2}} \qquad \text{B-10}$$

and for the closing cycle

$$\frac{dq}{d\theta} = K\left(\frac{1}{4N} - \theta\right)\frac{V-q}{V}\sqrt{\frac{V-q}{V} - \frac{L^2}{2\,gV\,h_o}\,\frac{d^2q}{d\theta^2}}$$

where

$$K = CL\pi RN\sqrt{2\,gh_o} \qquad \text{B-11}$$

The head at the beginning of the opening cycle is

$$h_o = \tfrac{1}{2} L\frac{\rho_{so} - \rho_I}{\rho} \qquad \text{B-12}$$

where

ρ_{so} = density of air in the door segment at the beginning of the opening cycle

ρ_I = density of air in room

ρ = reference density of air 0.075.

Fig. 2-B shows the air exchange rates for all four segments as calculated by equations B-10 and B-11 for assumed values of air temperature in the segment and indoors or outdoors. Note it does not represent the net infiltration but merely the amount of air entering or leaving the door segments.

In Equation B-12 the determination of ρ_{so}, the density of the air in the door segment, for a given indoor-outdoor temperature difference, requires a knowledge of the air temperature in the space. Thus in Fig. 3-B, t_{sc} is the air-space temperature just after closing. By inspection of the figure, equations can be written as

$$t_{so} = f_o\,t_o + (1 - f_o)\,t_{sc} \qquad \text{B-13}$$

and

$$t_{sc} = f_I\,t_I + (1 - f_I)\,t_{so} \qquad \text{B-14}$$

where

$$f_o = (\text{Air displacement as fraction of segment volume when segment is exposed to outdoors}) = \frac{q_o + q_c}{V}\ \text{outdoors} \qquad \text{B-15}$$

$$f_I = (\text{Air displacement as fraction of segment volume when segment is exposed to indoors}) = \frac{q_o + q_c}{V}\ \text{indoors} \qquad \text{B-16}$$

Combining equations B-13 and B-14

$$t_{sc} = \frac{f_I\,t_I + (1 - f_I)\,f_o\,t_o}{1 - (1 - f_o)\,(1 - f_I)} \qquad \text{B-17}$$

and

$$t_{so} = \frac{f_o\,t_o + (1 - f_o)\,f_I\,t_I}{1 - (1 - f_o)\,(1 - f_I)} \qquad \text{B-18}$$

The temperature of Segment I depends upon the relative quantity of indoor air m_I, and outdoor air, m_o in the segment.

$$t_{so} = \frac{t_I\,m_I + t_o\,m_o}{m_I + m_o} = \frac{t_I\,m_I + t_o\,m_o}{V\ (\text{volume of segment})} \qquad \text{B-19}$$

and similarly

$$t_{sc} = \frac{t_I\,m_{II} + t_o\,m'_o}{m_{II} + m'_o} = \frac{t_I\,m_{II} + t_o\,m'_o}{V} \qquad \text{B-20}$$

From Equations B-18 and B-19 the amount of indoor air in Segment I is

$$m_I = \frac{f_I\,(1 - f_o)\,V}{1 - (1 - f_o)\,(1 - f_I)} \qquad \text{B-21}$$

where for convenience, t_o was assumed to be zero.

Similarly from Equations B-17 and B-20 the amount of indoor air in Segment II is

$$m_{II} = \frac{f_I\,V}{1 - (1 - f_o)\,(1 - f_I)} \qquad \text{B-22}$$

The difference in the quantity of indoor air in the outgoing and incoming segments is the net infiltration. Hence,

Infiltration per segment per revolution $= m_{II} - m_I = \frac{f_o\,f_I\,V}{1 - (1 - f_o)\,(1 - f_I)}$ and the infiltration rate through the door becomes B-23

$$Q = \frac{4\,N\,f_I\,f_o\,V}{1 - (1 - f_o)\,(1 - f_I)},\ \text{cfm} \qquad \text{B-24}$$

In addition to knowing the infiltration through the revolving door the corresponding indoor-outdoor temperature difference must be found. By subtracting both sides of Equation B-18 from t_I it becomes

$$(t_I - t_{so}) = \frac{f_o}{1 - (1 - f_o)\,(1 - f_I)}\,(t_I - t_o) \qquad \text{B-25}$$

Also by subtracting t_o from both sides of Equation B-17 and dividing this result by Equation B-25, an equation is found which may help in visualizing some of the temperature and infiltration relationships.

$$\frac{t_{sc} - t_o}{t_I - t_{so}} = \frac{f_I}{f_o} \qquad \text{B-26}$$

Fig. 2-B shows air displacement rates for the entire door (i.e. 4 segments) and for (N) revolutions of the door as a function of ΔT. For a given ΔT the (f) values can be determined from Fig. 2-B with the following relationship:

$$f = \frac{\text{cfm from Fig. 2-B}}{4\,V\,N} \qquad \text{B-27}$$

where 4V was taken as 218 cu ft for the revolving door used in the tests.

For a given value of $t_I - t_{so}$, the net infiltration rate Q, and the corresponding indoor-outdoor temperature difference $(t_I - t_o)$ can be determined as follows:

1. Assume $t_i - t_{so}$ and obtain cfm (inside) from Fig. 2-B and determine f_i from Equation B-27.
2. Assume f_o. Where buoyance effects alone determine the infiltration, it is logical that $f_o = f_i$.
3. Knowing f_i and f_o, determine infiltration from Equation B-24 and $t_i - t_o$ from B-25.

Effect of turbulence and wind—By definition, the factors f_o and f_i represent the fractional air displacement of the segment volume when the segment is exposed to outdoors and indoors, respectively. This air displacement is caused by the temperature difference, in the absence of any external air turbulence or wind. Any external air movement will increase the value of f_o or f_i.

An approximate solution to the effect of air movement on infiltration was made by assuming that the average nondirectional air velocity head $\frac{v^2}{2g}$, was equivalent in its effect on infiltration to a buoyancy head of equal magnitude. For convenience, since curves of infiltration rates were plotted against the square root of the temperature difference between door segment air and room air, Fig. 2-B, the velocity heads were converted to temperature difference heads. Thus ΔT_T and ΔT_W were designated indoor turbulence and outdoor wind equivalent temperature heads.

To solve for the infiltration through the revolving door with a turbulence head inside of ΔT_T and wind outdoor equivalent to ΔT_W a trial and error solution was used as follows:

1. Assume $(t_i - t_{so})$ and ΔT_T.
2. Find cfm_i and thus (f_i) from Fig. 2-B for $\sqrt{(t_i - t_{so}) + \Delta T_T}$
3. Assume f_o.
4. Find $\sqrt{(t_{sc} - t_o) + \Delta T_W}$ from Fig. 2-B for assumed (f_o).
5. Calculate $(t_{sc} - t_o)$ from Equation B-26 if $[(t_{sc} - t_o)$ from Step 5$] > [(t_{sc} - t_o + \Delta T_W)$ from Step 4$]$ then a new value of (f_o) in Step 3 is required.
6. Solve for $\Delta T_W = [(t_{sc} - t_o) + \Delta T_W] - [t_{sc} - t_o]$.
7. Calculate $(t_i - t_o)$ from Equation B-25.
8. Calculate infiltration from Equation B-24.

From this procedure the indoor-outdoor temperature differential was found in Step 7, (reference t_o = zero) for the infiltration rate determined in Step 8, turbulence indoors of ΔT_T and wind outdoors of ΔT_W. In this manner the curves of Fig. 7 were constructed.

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1. ASHAE Research Report No. 1643, Winter Infiltration Through Swinging-Door Entrances in Multi-Story Buildings, by T. C. Min (ASHAE Transactions, Vol. 64, 1958, p. 421)
2. Infiltration Problem of Multiple Entrances by A. M. Simpson and K. B. Atkinson (ASHVE Journal Section, Heating, Piping & Air Conditioning, Vol. 8, No. 6, June, 1936, p. 345-351)

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Electronic analog computer solution of

Combined Heat and Mass Transfer problems

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Simultaneous transfer of mass and heat is a fundamental phenomenon of air conditioning. Cooling and dehumidifying coils, air washers, cooling towers and evaporative condensers are commonly used components in which combined heat and mass transfer occur; consequently, the ability to predict accurately the performance of these devices is an important and frequently recurring problem. This paper outlines methods by which these problems may be solved quickly and accurately on an electronic analog computer.

In a counter-flow or parallel-flow heat exchanger where mass transfer is not present (that is, only sensible heat transfer occurs) and where the flow rates, specific heats and over-all heat transfer coefficient are constant, the differential equations giving the air and refrigerant temperature distributions are integrated easily and result in the familiar equation $Q = US\Delta t_m$, where Δt_m is the over-all logarithmic mean temperature difference.

However, when mass transfer occurs, that is, when water vapor diffuses either from or to the air stream from the surface of the heat exchanger, the problem is complicated considerably. The heat transfer between the surface and the refrigerant must include not only the sensible heat transfer between the air and the surface, but the heat absorbed or released by the water vapor as it condenses or evaporates on the surface. Thus, a second driving force, the partial pressure or specific humidity gradient, and a second coefficient, the mass transfer coefficient, must be introduced to permit determination of the latent component of the total heat transfer.

Addition of these mass transfer terms makes it impossible to integrate the differential equations analytically, as is done easily in the case of sensible heat transfer only, because the surface temperature is no longer a linear function of heat transfer rate. Generally speaking, the solution to these problems is done in one step, assuming a logarithmic mean potential difference, with a trial and error or graphical method used to determine the surface temperature. The procedure is simplified by making the assumption that the Lewis number is unity, which permits utilization of an enthalpy potential for combined heat and mass transfer. However, for large cooling range, or if further complications are considered, such as the variation of heat transfer coefficient with coil depth, or variations in geometry, such as fins per in., several steps may be required in order to obtain an accurate solution. Obviously, such a calculation procedure is cumbersome and time consuming.

The great advantage of the electronic analog computer is that it can integrate the differential equations for the heat and mass transfer processes directly. Thus, within the limits of accuracy of the computer itself, exact solutions to these problems can be obtained. Furthermore, these solutions can be obtained much more quickly than by hand computation, especially for the more complex cases where a multistep solution is required. In the following sections the development of the analog solution will be detailed and its use illustrated by sample problems.

Basic equations – When moist air passes a surface whose temperature is below its dew point, heat is transferred from the air to the surface and, at the same time, water vapor diffuses to the cold surface, where it condenses. Consider a simple counterflow finned tube, as shown schematically in Fig. 1.

The sensible heat lost by the air in passing from x to x + dx on fin surface area dS_o is equal to the heat transferred to the fin surface,

$$dq_a = -w_a c_{pa} dt_a = \frac{(t_a - t_s) dS_o}{R_a} \tag{1}$$

where w_a is the air mass flow rate, c_{pa} is the specific heat of humid air, R_a is the air film thermal resist-

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ance based on air side surface area ($=1/h_a$), and t_a is the air dry bulb temperature.

Likewise, the water vapor leaving the air is equal to the water vapor diffusing to and condensing on the fin surface.

$$dtM = -w_a \, dW = k_m (W - W_s) \, dS_o = \frac{dtq_l}{\Delta h} \quad (2)$$

where Δh is the difference between the enthalpy of steam at the air dry bulb temperature and the enthalpy of liquid water at the fin surface temperature, k_m is the mass transfer coefficient and W is the specific humidity of the air.

Finally, the heat which is transferred through the fin and tube to the sink fluid equals the heat gained by the sink fluid,

$$dtq = \frac{(t_s - t_l) \, dS_o}{R_m + R_l} = \begin{array}{ll} -w_l \, c_{pl} \, dt_l & \text{for counterflow} \\ +w_l \, c_{pl} \, dt_l & \text{for parallel flow} \end{array} \quad (3)$$

where w_l and c_{pl} are the mass flow rate and specific heat of the sink fluid, respectively, R_l is the sink fluid thermal resistance based on air side surface area and R_m is the thermal resistance of the fin and tube material based on air side surface area. As shown by Carrier and Anderson,[1] the term R_m is substantially constant over a fairly wide range of heat transfer coefficients for a given fin conductivity and geometry.

Equations (1) through (3) are the three basic differential equations describing the heat and mass transfer processes. They may be rewritten in a form more convenient for application by dividing both sides by the air side frontal area, A, and expressing in integral form,

$$t_a - t_{ao} = -\int_0^x \frac{s_v}{G_a \, c_{pa}} \left[\frac{t_a - t_s}{R_a} \right] dx \quad (4)$$

$$W - W_o = -\int_0^x \frac{s_v}{G_a} k_m (W - W_s) \, dx \quad (5)$$

$$t_l - t_{lo} = \pm \int_0^x \frac{s_v}{G_l \, c_{pl}} \left(\frac{A}{A_l} \right) \left[\frac{t_s - t_l}{R_m + R_l} \right] dx \quad (6)$$

where s_v is the air side surface area per unit total core volume, or compactness, G_a is the air mass frontal velocity, G_l is the sink fluid mass frontal velocity and A/A_l is the ratio of the frontal flow cross sectional areas. The subscript o refers to conditions at x = o. The minus sign in Equation (6) applies to counterflow and the plus sign to parallel flow operation.

Equations (4) through (6) are three integral equations with five dependent variables, t_a, t_s, t_l, W and W_s. To obtain solutions, two more equations are required. The first is simply the relationship between the specific humidity of saturated air and temperature, which applies at the fin surface.

$$W_s = f(t_s) \quad (7)$$

Tables of the thermodynamic properties of moist air may be found in Reference 2.

The second relationship is obtained by equating the total heat transferred to the sink fluid to the sum of the latent and sensible components transferred to the fin, i.e., the first law of thermodynamics. Thus, combining Equations (1) through (3) gives

$$\frac{t_a - t_s}{R_a} + k_m \Delta h (W - W_s) = \frac{t_s - t_l}{R_m + R_l} \quad (8)$$

Equations (4) through (8) are five independent equations which must be solved simultaneously for the five dependent variables as functions of the independent variable x.

Note that these equations, though derived for cooling and dehumidifying, are valid for any combination of heating or cooling and humidification or dehumidification, provided the surface is wet. For dry surface, however, the mass transfer term in Equation (8) must be eliminated and Equation (5) does not apply.

The solution to Equations (5) and (8) requires knowledge of the mass transfer coefficient, k_m. Generally, such data are not available. However, due to the similarity between the heat and mass transfer process, the mass transfer coeffi-

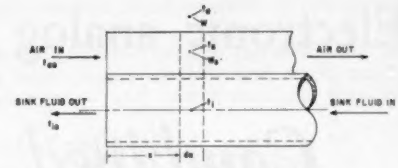


Fig. 1 Schematic of counterflow extended surface heat exchanger

cient may be related to the heat transfer coefficient, which we assume is known, through the Lewis Relation.*

$$k_m = \frac{1}{R_a \, c_{pa} \, N_{Le}} \quad (9)$$

Substitution of Equation (9) into Equations (5) and (8) permits determination of the heat exchanger performance, given only heat transfer data.

Electronic Analog Computer Solution – Two schematic computer network diagrams for the solution to Equations (4) through (9) are shown in Figs. 2 and 3. The first, and simplest, provides the solution for the case where the parameters s_v, G_a, A/A_l, G_l, c_{pl}, N_{Le}, Δh, R_a, R_m and R_l are each assumed constant with respect to heat exchanger depth. In this case only amplifiers, potentiometers and one diode function generator are required for a solution. The second and more general analog solution assumes that G_a, c_{pa}, A/A_l, G_l, c_{pl}, N_{Le} and Δh are constant, but permits an arbitrary variation of s_v, R_a and $R_m + R_l$ with coil depth. This requires three multipliers and three diode function generators in addition to the amplifiers and potentiometers. Of course, if more multipliers and function generators are available, it would not be necessary to assume any of these parameters constant.

Referring to the circuit diagrams, it will be seen that in both cases, amplifier 1 integrates Equation (4) for air dry bulb temperature, amplifier 2 integrates Equation (5), giving specific humidity, amplifier 3 integrates Equation (6), solving for the sink fluid temperature, and amplifier 4 solves Equation (8). The relationship between specific humidity and temperature at saturation is programmed into

* Note that N_{Le} as used here is defined as in Reference 3, as $(\alpha/\delta)^{2/3}$. The Lewis number also is defined commonly as α/δ.

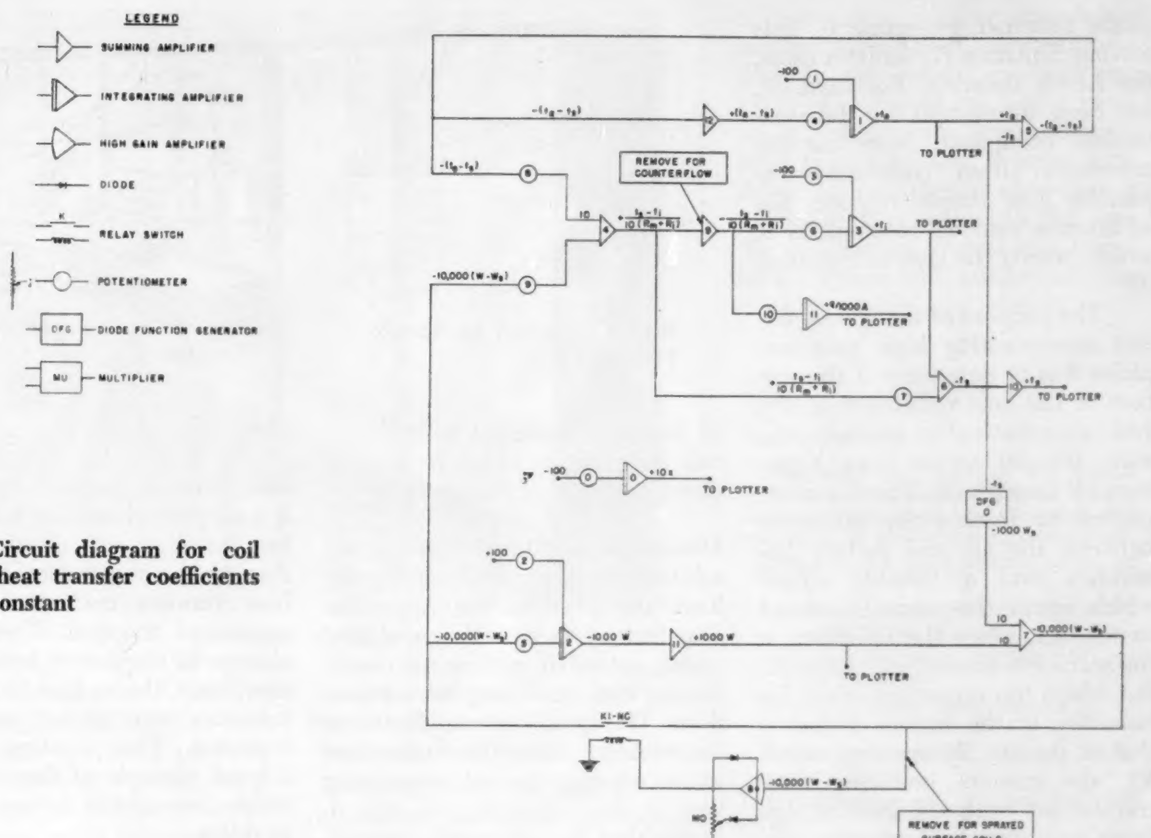


Fig. 2 Circuit diagram for coil problem; heat transfer coefficients assumed constant

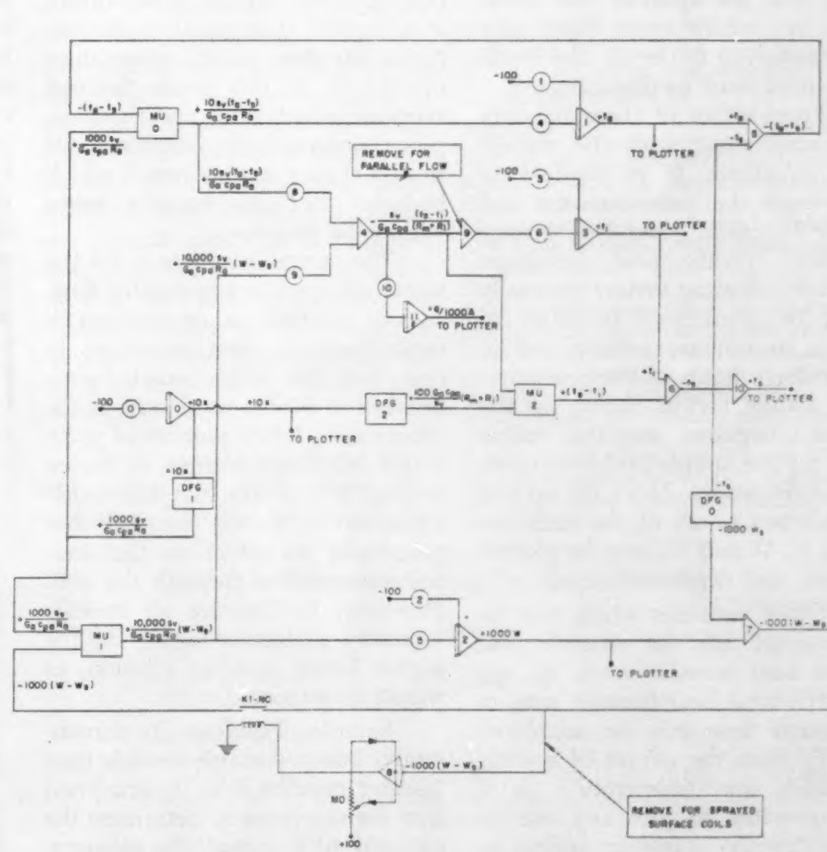


Fig. 3 Circuit diagram for coil problem; heat transfer coefficients functions of coil location

diode function generator 0, thus solving Equation (7). In both cases, the Lewis Relation, Equation (9), has been substituted for the mass transfer coefficient. Note that the conversion from counterflow to parallel flow simply requires the addition or removal of amplifier 9, which inverts the sign in Equation (6).

The purpose of the special circuit incorporating high gain amplifier 8 is to determine if the surface of the heat exchanger is dry, and automatically exclude the mass transfer terms from Equations (4) through (8). This is accomplished by feeding the difference between the air and surface humidities into a bistable circuit which opens the normally closed switch, K1, when the humidity at the surface is above that of the air, but which has no output when the humidity at the surface is below that of the air. By opening switch K1, the circuits involving mass transfer are rendered inactive, but those involving heat transfer continue to function. Heat exchangers which operate partly dry and partly wet thus are analyzed easily. Note that for sprayed coil problems, i.e., where water vapor may be transferred to the air, the bistable circuit must be disconnected.

From either of these network diagrams, along with the respective equations, it is possible to determine the potentiometer and diode function generator settings for any specific heat exchanger problem. Scaling factors generally must be introduced in order to obtain maximum accuracy within the voltage limits of the computer. The scaling factors shown on the circuit diagrams are the values used for the sample problems given in the Appendix. Once set up and scaled, any or all of the variables t_s, t_a, t_i, W and W_s may be plotted against coil depth, x.

Other variables which may be of interest are the sensible and latent heat transfer rates, q_s, q_l, and the total heat transfer rate, q. Although these may be calculated readily from the curves of specific humidity and temperature, it is also possible to plot any one of them directly, simply by adding an integrator. For example, since the value of the total heat transfer rate is almost always desired, integrator

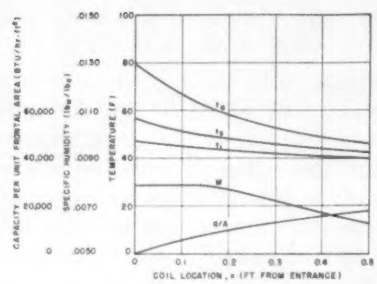


Fig. 4 Solution to Sample Problem 1

11 has been included in both circuit diagrams in order to provide direct read-out of this quantity.

Discussion – Detailed computer solutions for three illustrative problems are given in the Appendix. The first of these is for a chilled water, extended surface air conditioning coil operating in counterflow. This problem was chosen to demonstrate how the computer senses whether the coil is operating wet or dry. Referring to Fig. 4, notice that the air specific humidity is constant for the first 0.12 ft of coil depth. This portion of the coil is operating dry. Also notice that the coil surface temperature at x = 0.12 ft is equal to the entering air dew point temperature of 50.5 F. At this point, the coil starts dehumidifying, and the computer automatically accounts for this by closing a relay switch which includes the mass transfer terms within the problem.

The second problem is for the same coil operating in parallel flow. In this problem a comparison is made between two solutions; in one case the more exact Lewis number of 0.87 is used, and in the other case a Lewis number of unity is utilized. The solutions, as shown in Fig. 5, indicate that this small variation in Lewis number has practically no effect on the temperature profiles through the coil. However, the leaving air specific humidity is slightly higher for the higher Lewis number solution, as would be expected.

Sample Problem 3 demonstrates how a variable air side heat transfer coefficient is programmed into the computer to determine the capacity of a prime tube evaporative condenser. Fig. 6 shows the plotted solution for this problem. In this figure the surface tempera-

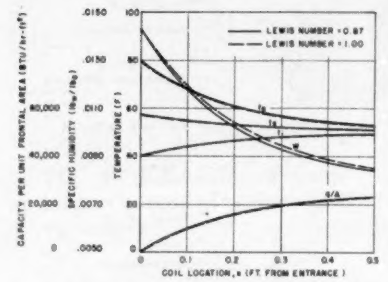


Fig. 5 Solution to Sample Problem 2

ture curve best illustrates the effect of the variable heat transfer coefficient. Notice that the tube surface temperature decreases for the first two rows of coil depth. This is due to the rapid increase of the heat transfer coefficient for this section of the coil. Then, as the change in coefficient becomes less significant, the surface temperature increases with air temperature, as expected. This problem serves as a good example of the computer's ability to solve a rather complex problem.

The authors have found the analog computer method of solving simultaneous heat and mass transfer coil problems quite useful and time saving. Its accuracy is largely dependent on the read-out mechanism of the computer; however, with a good machine and the problem properly scaled, the accuracy is better than most hand computational methods. Given the necessary coil parameters, the time required to solve problems of this type using the computer will normally be about one hr. However, when a series of similar problems must be solved, ten or more solutions may be obtained with little extra effort. Also, since the com-

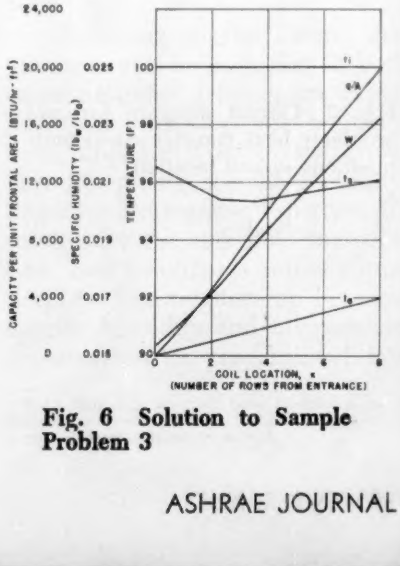


Fig. 6 Solution to Sample Problem 3

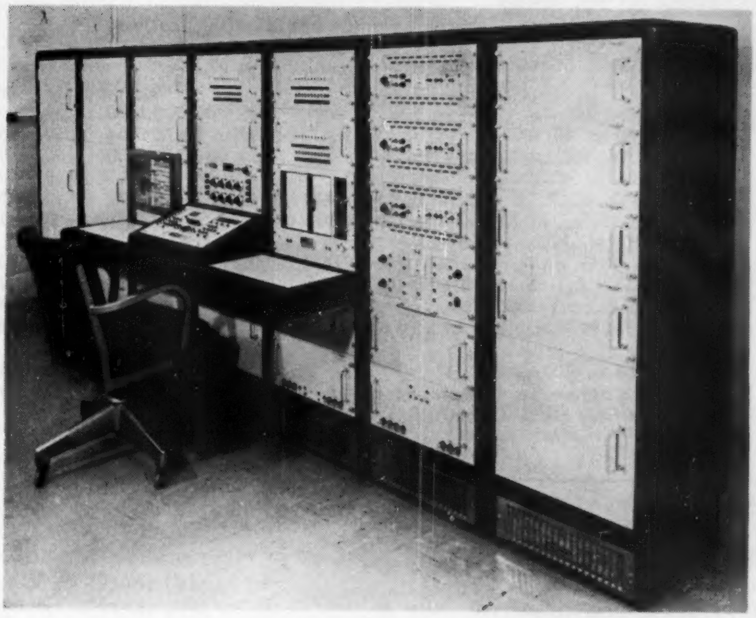


Fig. 7 Electronic analog computer

puter solution is a plot of coil performance versus coil location, each plot gives the operating performance for a series of various depth coils operating at the same conditions. Caution must be used here, though, since heat transfer coefficients may not be equal for different depth coils. This is especially true for the air side coefficient of shallow coils. This problem may be eliminated, however, by programming the variation of these coefficients into the computer, as was done in Sample Problem 3.

Although the equations developed in this report apply to coil problems, the method itself may be extended to a number of systems involving heat or mass transfer. Also, since it is unnecessary to make the assumption that the Lewis number equals unity, this method is not restricted to air and water mixture systems.

Summary – The differential equations describing the combined heat and mass transfer from air to an extended surface counterflow or parallel flow heat exchanger have been derived in terms convenient for solution with an electronic analog computer. The specific humidity difference between the air and the fin surface is taken as the driving force for mass transfer, and the mass transfer coefficient is related to the heat transfer coefficient through the Lewis Relation. Whereas it is customary to use the enthalpy of moist air as the potential for combined heat and mass transfer, this assumption is not necessary for the computer solution.

Two circuit diagrams for computer solutions to this problem are presented. The first, and simplest – which assumes constant heat transfer coefficients, flow rates and specific heats – gives solutions of satisfactory accuracy for most air conditioning problems. The second, and more general, solution allows inclusion of a variation in heat transfer coefficient on either or both sides with flow length. In the case of a cooling and dehumidifying heat exchanger, which may operate partly dry, a special circuit is incorporated which senses whether the surface temperature is above or below the air dew point temperature and automatically includes or excludes the mass transfer terms.

Three sample problems have been solved to demonstrate the technique. The first problem is for a counterflow cooling and dehumidifying coil which operates partly dry. In the second problem the effect of changing the Lewis number is shown for the same coil operating in parallel flow. A prime tube evaporative condenser was selected for the third problem. In this case the variation in air side heat transfer coefficient with number of tube rows was taken into account.

APPENDIX

Sample Problem 1

Determine the leaving air conditions of a counterflow, chilled water, extended surface air conditioning coil operating at the following conditions:

Entering air temperature $= t_{ao} =$ 80 F

Entering air specific humidity $= W_o$ $= 0.0079$ lb_w/lb_a

Entering air mass velocity $= G_a =$ 1800 lb_a/hr-ft^2

Entering water temperature $= t_{li}$ $= 40$ F

Water mass velocity $= G_l = 1{,}200{,}000$ lb/hr-ft^2

The following coil parameters are given:

$x = 0.5$ ft

$A/A_l = 500$

$s_v = 300$ ft^2/ft^3

$R_a = 0.100$ hr-ft^2-F/Btu

$R_m = 0.016$ hr-ft^2-F/Btu

$R_l = 0.025$ hr-ft^2-F/Btu

$N_{Le} = 0.87$ (good for most air conditioning temperatures, see Reference 3)

Solution

First, it is necessary to tabulate equations for the potentiometer settings and calculate the respective numerical values as shown in Table I. The value 10 found in the denominator of potentiometer Equations (4), (5), (6) and (10) is a scaling factor that must be used since one ft of coil depth has been set equal to ten sec of machine time. Next, the diode function generator settings are tabulated. Values of the saturated specific humidity of air at standard atmospheric pressure are listed in Table II as a function of surface temperature. With this information tabulated, the computer is wired according to the network diagram of Fig. 2, with the

Table I Potentiometer Setting For Sample Problem I

Potentiometer	Equation	Numerical Value
0	$1/10$	0.1000
1	$t_{ao}/100 = 80/100$	0.8000
2	$1000\ W_o/100 = 10(0.0079)$	0.0790
3	$t_{lo}/100 \cong 50/100$	0.5000
4	$s_v/10\ G_a\ c_{pa}\ R_a = 300/10(1800)\ (0.244)\ (0.10)$	0.6830
5	$s_v/100\ G_a\ c_{pa}\ R_a\ N_{Le} = 300/100(1800)\ (0.244)\ (0.10)\ (0.87)$	0.0785
6	$10 s_v\ A/10\ G_l\ c_{pl}\ A_l = 10(300)\ (500)/10(1{,}200{,}000)\ (1.0)$	0.1250
7	$10(R_m + R_l) = 10(0.016 + 0.025)$	0.4100
8	$1/100\ R_a = 1/100(0.10)$	0.1000
9	$\Delta h/100{,}000\ R_a\ c_{pa}\ N_{Le} = 1061/100{,}000(0.10)\ (0.244)\ (0.87)$	0.5000
10	$s_v/1000 = 300/1000$	0.3000
MO	Setting determined by voltage requirement of holding coil in relay switch KI (50 to 70 volt).	0.6300

Table II Setting Of Diode Function Generator O

Surface Temperature t_s		Specific Humidity $1000\ W_s$*
0		0.787
5		1.020
10		1.315
15		1.687
20		2.152
25		2.733
30		3.454
35	Range Programmed for Sample Problems 1 and 2	4.275
40		5.213
45		6.331
50		7.658
55		9.229
60		11.08
65		13.26
70		15.82
75		18.82
80		22.33
85	Range Programmed for Sample Problem 3	26.42
90		31.18
95		36.73
100		43.19

* Values taken from Reference 2.

Table III Potentiometer Settings For Sample Problem 2

Potentiometer	Equation	Numerical Value $N_{Le} = 0.87$	Numerical Value $N_{Le} = 1.00$
0	$1/10$	0.1000	0.1000
1	$t_{ao}/100 = 80/100$	0.8000	0.8000
2	$1000\ W_o/100 = 10(0.0143)$	0.1430	0.1430
3	$t_{1o}/100 = 40/100$	0.4000	0.4000
4	$s_v/10\ G_a\ c_{pa}\ R_a = 300/10(1800)\ (0.244)\ (0.10)$	0.6830	0.6830
5	$s_v/100\ G_a\ c_{pa}\ R_a\ N_{Le} = 300/100(1800)\ (0.244)\ (0.10)\ (N_{Le})$	0.0785	0.0683
6	$10 s_v A/10\ G_1\ c_{p1}\ A_1 = 10(300)\ (500)/10(1{,}200{,}000)\ (1.0)$	0.1250	0.1250
7	$10(R_m + R_1) = 10(0.016 + 0.025)$	0.4100	0.4100
8	$1/100\ R_a = 1/100(0.10)$	0.1000	0.1000
9	$\Delta h/100{,}000\ R_a\ c_{pa}\ N_{Le} = 1061/100{,}000(0.103\ (0.244)\ (N_{Le})$	0.5000	0.4350
10	$s_v/1000 = 300/1000$	0.3000	0.3000
MO	Setting determined by voltage requirement of holding coil in relay switch K1 (50-70 volt)	0.6300	0.6300

exception that amplifier 9 is omitted, since operation is counterflow. Finally, the diode function generator and potentiometers are set and the solution is obtained by plotting the variables t_a, W, t_1 and t_s against coil location x.

Note that the value calculated for potentiometer 3 is but an approximate setting, since the leaving water temperature is unknown. However, by a manual trial and error procedure this potentiometer setting is varied until the entering water temperature (t_1 at $x = 0.5$ ft) is equal to 40 F. Fig. 4 shows the plotted solution for this problem. The leaving air conditions (t_a and W at $x = 0.5$) are 46.0 F and 0.0063 lb_w/lb_a.

Sample Problem 2

For the cooling coil of Problem 1, determine the leaving air conditions when the coil is circuited for parallel flow and operating at the following conditions:

- Entering air temperature $= t_{ao} =$ 80 F
- Entering air specific humidity $= W_o = 0.0143\ lb_w/lb_a$
- Entering air mass velocity $= G_a =$ 1800 lb_a/hr-ft^2
- Entering water temperature $= t_{1o} = 40$ F
- Water mass velocity $= G_1 = 1{,}200{,}000$ lb/hr-ft^2

Also, determine the leaving air conditions for this coil if a Lewis number of unity is assumed.

Solution

First, calculate the potentiometer values and list them in tabular form as shown in Table III. Then, tabulate the diode function generator settings. These settings will be the same as shown in Table II for Problem 1. Next, the computer is wired according to Fig. 2, and the diode function generator and potentiometers are set. Plot the variables t_a, W, t_s and t_1 against coil location x. Then, reset potentiometers 5 and 9 to 0.0683 and 0.4350, respectively, and replot the variables for the Lewis number of unity solution. These plotted solutions appear in Fig. 5, which shows the leaving air humidity to be 0.0084 lb_w/lb_a for a Lewis number of 0.87 and 0.0085 lb_w/lb_a for a Lewis number of 1.00. The leaving dry bulb temperature is about 53.0 F for each case.

NOMENCLATURE

Symbol	Concept	Typical Units
α	Thermal diffusivity	(ft^2) (hr)$^{-1}$
δ	Mechanical diffusivity	(ft^2) (hr)$^{-1}$
A	Air frontal flow cross sectional area	ft^2
A_1	Sink fluid frontal flow cross sectional area	ft^2
c_{pa}	Constant pressure specific heat of moist air	(Btu) $(lb_a)^{-1}$ (F)$^{-1}$
c_{p1}	Constant pressure specific heat of sink fluid	(Btu) (lb)$^{-1}$ (F)$^{-1}$
G_a	Mass frontal velocity of air	(lb) (hr)$^{-1}$ (ft^2)$^{-1}$
G_1	Mass frontal velocity of sink fluid	(lb) (hr)$^{-1}$ (ft^2)$^{-1}$
Δh	Enthalpy difference between water vapor at air temperature and liquid water at plate surface temperature	(Btu) (lb)$^{-1}$
h_a, h_{ao}	Air side heat transfer coefficient	(Btu) (hr)$^{-1}$ (ft^2)$^{-1}$ (F)$^{-1}$
k_m	Mass transfer coefficient	(lb_w) (hr)$^{-1}$ (ft^2)$^{-1}$ $(lb_w/lb_a)^{-1}$
M	Mass transfer flow rate	(lb_w) (hr)$^{-1}$
N_{Le}	Lewis number $= (\alpha/\delta)^{2/3}$	———
q_1	Latent heat flow rate	(Btu) (hr)$^{-1}$
q_s	Sensible heat flow rate	(Btu) (hr)$^{-1}$
q	Total heat flow rate	(Btu) (hr)$^{-1}$
R_a	Thermal resistance of air	(hr) (ft^2) (F) (Btu)$^{-1}$
R_1	Thermal resistance of sink fluid	(hr) (ft^2) (F) (Btu)$^{-1}$
R_m	Thermal resistance of metal	(hr) (ft^2) (F) (Btu)$^{-1}$
R_s	Ratio of air side to tube inside surface area	———
S	Surface area	ft^2
S_o	Outside surface area	ft^2
s_v	Outside surface area per unit core volume	(ft^2) (ft^3)$^{-1}$, (ft^2) (ft^2)$^{-1}$ (row)$^{-1}$
t_a	Air temperature	F
t_1	Refrigerant or sink fluid temperature	F
t_s	Coil air side surface temperature	F
Δt_m	Logarithmic mean temperature difference	F
U	Over-all heat transfer coefficient	(Btu) (hr)$^{-1}$ (ft^2)$^{-1}$ (F)$^{-1}$
w_a	Dry air flow rate	(lb_a) (hr)$^{-1}$
w_1	Sink fluid flow rate	(lb) (hr)$^{-1}$
W	Specific humidity of air	(lb_w) $(lb_a)^{-1}$
W_s	Specific humidity of saturated air at coil surface temperature	(lb_w) $(lb_a)^{-1}$
x	Coil location (distance from entrance)	ft, rows

Sample Problem 3

Determine the capacity of a prime tube evaporative condensing coil having a variable air side heat transfer coefficient dependent on coil location x.

Table IV Potentiometer Settings For Sample Problem 3

Potentiometer	Equation	Numerical Value
0	1/10	0.1000
1	$t_{ao}/100 = 90/100$	0.9000
2	$1000\ W_o/100 = 10(0.0153)$	0.1530
3	$t_{1o}/100 = 100/100$	1.000
4	1/10	0.1000
5	$1/10\ N_{Le} = 1/10(0.87)$	0.1150
6	Not used	——
7	$(R_m + R_t)\ G_a\ c_{pa}/s_v = (0.0027)\ (2250)\ (0.244)/1.594$	0.9300
8	1/10	0.1000
9	$\Delta h/10{,}000\ c_{pa}\ N_{Le} = 1061/10{,}000\ (0.244)\ (0.87)$	0.5000
10	$G_a\ c_{pa}/1000 = (2250)\ (0.244)/1000$	0.5490
MO	Setting determined by voltage requirement of holding coil in relay switch Kl (50 to 70 volt).	0.6300

Table V Settings of Diode Function Generator I for Sample Problem 3

10x	h_a/h_∞*	$1000\ s_v/G_a\ c_{pa}\ R_a$
0	0.565**	30.40
5	0.630	33.90
15	0.760	40.90
25	0.930	50.00
35	0.980	52.70
45	0.990	53.30
55	1.000	53.80
65	1.000	53.80
75	1.000	53.80
85	1.000	53.80

* These values taken from Reference 4.
** This value extrapolated.

This coil will operate at the following conditions:

Entering air temperature $= t_{ao} =$ 90 F

Entering air specific humidity $= W_o = 0.0153$ lb_w/lb_a

Entering air mass velocity $= G_a =$ 2250 lb_a/hr-ft²

Condensing temperature $= t_t = 100$ F

The following coil parameters are given:

$x = 8$ rows

$s_v = 1.594$ ft²/ft²-row

$A = 10$ ft²

$R_m \approx 0$

$R_t = 0.0027$ hr-ft²-F/Btu

$h_\infty = 18.55$ Btu/hr-ft²-F

$R_a = 1/h_a = 1/[h_\infty\ (h_a/h_\infty)] =$

$h_a/h_\infty = 0.0539/(h_a/h_\infty)$ hr-ft²-F/Btu

f (x) Table V lists the ratio h_a/h_∞ as a function of coil location x.

Solution

Fig. 3 shows the computer network diagram required when the terms R_a, R_m, R_t and s_v are variable. In this problem the term R_a is a variable while the terms R_m, R_t and s_v are constants. Because of this, diode function generator 2 and multiplier 2 shown in Fig. 3 may be replaced by potentiometer 7 wired as shown in Fig. 2. Also, since condensing temperature t_t is constant, integrating amplifier 3 may be replaced by a summing amplifier such that potentiometer 3 feeds directly into amplifier 3 and potentiometer 6 is omitted from the circuit.

With the above modifications to the network diagram, the potentiometer and diode function generator settings are calculated as shown in Tables II, IV and V. Finally, the computer is wired and the potentiometers and diode function generators are set. Fig. 6 shows the values t_a, W, t_s and q/A plotted against coil location x. This plot shows the capacity of an eight-row coil having a face area of ten sq ft to be 199,000 Btu/hr.

Note that the curves given in Fig. 6 may also be used to determine the capacity of coils of other depths since the proper variation in air side heat transfer coefficient with coil depth has been programmed. For example, the capacity of a four-row coil of ten sq ft face area would be 100,000 Btu/hr, other things being equal.

REFERENCES

1. Carrier, W. H. and Anderson, S. W., "The Resistance to Heat Flow Through Finned Tubing", ASHVE Transactions, Vol. 50, 1944, p. 117.
2. ASHRAE GUIDE, 38th Edition, 1960.
3. ASRE DATA BOOK, 10th Edition, 1957-58.
4. McAdams, W. H., Heat Transmission, McGraw-Hill Book Company, Inc., New York, 1954.

BULLETINS

Pumps. Design and construction features of Electri-Cand pumps are presented in four-page Bulletin 52B-8471B. Completely sealed, the units utilize pumped liquids flowing through the motor to serve as a cooling agent and bearing lubricant. Pumps are available in sizes from 1 x ¾ to 3 x 3 in., with capacities to 600 gpm, heads to 275 ft and pressures to 200 psi at temperatures to 200 F with standard insulation and 400 F with special insulations. Range of hp is from ½ through 25.
Allis-Chalmers, Manufacturing Company, Milwaukee 1, Wisc.

Fans. Sirocco fans, Series 81, Class I and II, are the subject of 48-page Bulletin A-1401. Units are discussed in terms of their construction features and operating characteristics. Drive arrangements, control methods and types of drive are described and illustrated, together with specific component features such as bearings and sound-absorbing bases. Capacity tables are provided for the full line of 15 sizes, single-inlet single-width design, and 12 sizes, double-inlet double-width design. Outline type drawings are keyed to tabular data to provide important dimensions.
American-Standard, Industrial Div, Detroit 32, Mich.

Construction Material. Combining the physical properties of a fiber-base laminate with the physical and chemical properties of rigid polyvinyl chloride, Kaylite is the subject of a series of data sheets. In addition to the chemical resistance of rigid PVC, the material may be exposed to higher and lower temperatures than recommended for rigid PVC, due to the fact that structural strength is provided by the fiber-base laminate. Covered in the data sheets are physical and chemical properties, uses and applications (such as in air conditioning ducts) and fabrication.
Kaykor Products Corporation, Yardville, N. J.

Heating and Ventilating Units. Contained in a comprehensive engineering catalog for large heating and ventilating units and heat diffusers are selection charts, steam and hot water coil capacities, fan-motor ratings and dimensional data for all models and sizes produced by this manufacturer. These units, which are designed for commercial and other large buildings, provide heating from 47,000 to 2,900,000 Btu/hr with air capacities of 1500 to 32,000 cfm.
Carrier Air Conditioning Company, Syracuse 1, N. Y.

Tubular Products. Coils, bends and fabricated piping available from this manufacturer for numerous heating applications are listed in four-page Bulletin PC-61. Pipe coils can be fabricated from any ferrous or nonferrous metal, in any shape or design, and engineered to specific heat absorption or radiation requirements.
Rempe Company, 340 N. Sacramento Blvd., Chicago 12, Ill.

Insulation System. An integrated pipe insulation that combines metal jacketing, high temperature calcium silicate insulation and moisture barrier in one factory-fabricated unit is detailed in eight-page Bulletin IN-217A. Describing and illustrating the Metal-On System, the bulletin enumerates its advantages and covers specific installations.
Johns-Manville, 22 E. 40th St., New York 16, N. Y.

The what and how of *Research Proposals*

S. F. GILMAN
Chairman
Research and Technical Committee

Many of the Technical Committees and Task Groups now have their desired complement of members, and several TCs are actively studying their research needs. It is heartening to learn that some are already drafting research proposals, while others are evaluating proposals already submitted by research institutions.

As further encouragement, Martin A. Mayers has been appointed Manager of Research (see page 90, this issue), thus making some Headquarters assistance available to the TCs during this period of rapidly increasing activity. Future research pages will tell how this assistance can be most readily invoked.

We now have research capabilities at Society Headquarters and in many TCs and Task Groups. How then do they work with research institutions to obtain research results of maximum benefit to the Society membership?

First of all, a research need must be established; the determination of this need is the basic responsibility of each TC in the field of its interest. Once such a need is established the TC may outline the work it wishes to have done and ask the Manager of Research to find a place to have it done. If the TC knows a cooperating institution able and willing to perform the work it may help the institution to prepare a proposal and offer the complete proposal with its endorsement to the R & T Committee.

Research institutions may themselves play an important role in the process, for they may already be aware of a research need, have a qualified staff member available and only need the funds to get going. In such a case, an institution should contact the Chairman of the cognizant TC and work with him in preparing a proposal. If there is a question as to which of the 69 TCs and 6 Task Groups (See page 76 of the September JOURNAL) is most appropriate for the work, Manager of Research Martin A. Mayers should be contacted at ASHRAE, 345 East 47th Street, New York 17, N. Y.

The research proposal need not be elaborate but the objective and scope of the proposed research should be quite specific. It should include at least the following information:

1. Statement of the problem.
2. Objectives and scope of proposed research.
3. Background (availability of special talents, facilities or instruments in technical field of problem; ultimate benefit of ASHRAE members; etc.)
4. Proposed method of solving problem.
5. Proposed time schedule and budget.

To minimize the work required in formulating proposals, we plan to develop and make available a Research Proposal Form. This form will have the essential topic items printed on it and space will be provided between topics for typing the essential information. This, together with an Instruction Sheet, should simplify the task of writing up research proposals. Also, the Manager of Research will actively assist TCs and research institutions in the details of proposal formulation and processing.

The procedure for processing Research Proposals is straightforward. The TC or Task Group approves the proposed project and recommends it to the Research and Technical Committee. The Section Head is a member of the R & T Committee and furnishes supplementary information as required during the deliberations of the Committee. If the R & T Committee approves the proposal, final action is taken by the Board of Directors, which is the body having authority to commit Society funds.

Note that the initial step in the procedure is a favorable recommendation from a TC or Task Group. It is here that the detailed qualification of a proposal is conducted. Thus, any Research Proposals submitted directly to Society Headquarters is assigned to a TC or Task Group for qualification. If the proposal falls outside the scope of a single TC, a Task Group can be established for the purpose of conducting the evaluation.

The Manager of Research then negotiates a contract with the research institution, the details of which are subject to the approval of the Executive Secretary and the President. A contract is ordinarily made for one year ending June 30, but is easily extended, if mutually desired, by the Society's writing of a brief Letter of Extension.

Financial support of research projects can be provided by various ways. If the research is potentially of benefit to a large segment of the membership, the so called "general fund" can be used. This fund is comprised mainly of the contribution to the Society from the biennial Exposition and solicited contributions from industry which are not restricted to specific research projects or technical areas by the contributor.

If a research project will benefit a specific industry group, then it is expected that financial support will be obtained by solicitation of that group. However, the R & T Committee realizes that finincial support is generally more easily obtained for an active project than for one in the proposal stage. Consequently, as the need arises general funds are allocated to "seed" particular projects and get them active. In due course, responsibility for continuing financial support goes to the industry group intended.

Universities having active projects or research proposals currently in process are:

Arizona	Kansas State
Case Institute	Kentucky
Columbia	Northwestern
Florida	U C L A
Illinois	Wisconsin

Several other potential research needs are currently being qualified by TCs, and some TCs and Task Groups are working with universities to determine if research proposals should be formulated.

Progress to date has been most encouraging, and the R & T Committee is optimistic that the cooperative research activity will attain the desired level within the next several months. Meeting the goal that has been set will require continuing the hard work and fine cooperation that have been so conclusively demonstrated by the Technical Committees and Task Groups during the year so far.

New development in

Steam Vacuum Refrigeration

ELLIOT SPENCER
Associate ASHRAE

Steam vacuum refrigeration was first investigated some time prior to 1901 by LeBlanc and Parsons, and was applied quite limitedly at that time. Limitations of early ejector design and the primitive nature of pumps and controls, however, prevented its widespread use.

Despite these shortcomings, steam vacuum refrigeration systems were applied successfully during the early stages of our industry, and some of these systems are still in operation after 20 or 30 years of reliable service. During the past 25 years, improvements in accessory design and ejector efficiency have resulted in the application of these systems in increasing numbers for industrial and commercial water chilling requirements.

Regardless of the many known advantages of the conventional steam vacuum refrigeration cycle, which include reliability (there are no moving parts), low maintenance, the use of water exclusively as a refrigerant and the ability to use either high or low pressure steam, it was not applied as extensively to comfort air conditioning as other systems, such as centrifugal refrigeration and lithium bromide absorption. The explanation of this situation lies in the basic characteristics of steam vacuum refrigeration.

Simply stated, as the available condenser water temperature rises, both the steam rate (lb of steam per hr per ton of refrigeration) and the condenser water rate (gpm per ton of refrigeration) increase. This means that unless large quantities of condenser water are available at low cost, as from a river or other large body of water, or unless water is available at a low temperature, as from a well, cooling water requirements will exceed substantially that of the other refrigeration systems. When water from a cooling tower is used at the conventional 85 F level, steam vacuum refrigeration will have an indicated full load steam rate and water rate which is somewhat higher than either centrifugal refrigeration or lithium bromide absorption.

Typical conditions are tabulated herewith. Thus, when river or sea water is available, or some other source of cool water, steam vacuum refrigeration systems are equal, or superior, to other systems insofar as steam and water consumption are concerned. First cost is often lower. When 85 F cooling tower water is used for comfort air conditioning, chilled water temperatures of 45 to 50 F, steam vacuum refrigeration usually is ruled out. At higher chilled water temperatures, steam vacuum refrigeration will be more competitive from a utility standpoint. This can be understood better by reference to the performance characteristics of steam vacuum refrigeration systems which are tabulated and illustrated graphically herein (Fig. 5). (See Note 1.)

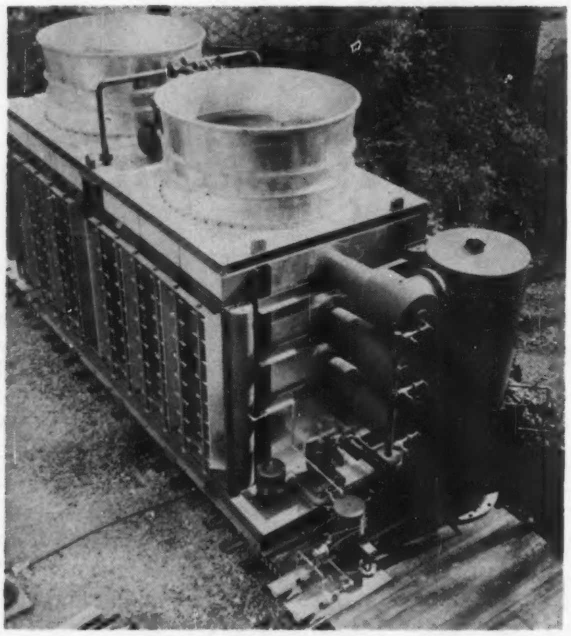
Obviously, any modification of the steam vacuum refrigeration cycle, which reduces its steam consumption and eliminates the need for condensing water in large quantities, will greatly affect the feasibility and usefulness of steam vacuum refrigeration for comfort air conditioning applications.

It is this modification which concerns us. By the simple expedient of replacing the water-cooled condenser of conventional steam vacuum refrigeration with a wetted air-cooled surface condenser, steam consumption is reduced and water requirements are lowered to fractional levels. Large condenser water recirculating pumps are, of course, completely eliminated.

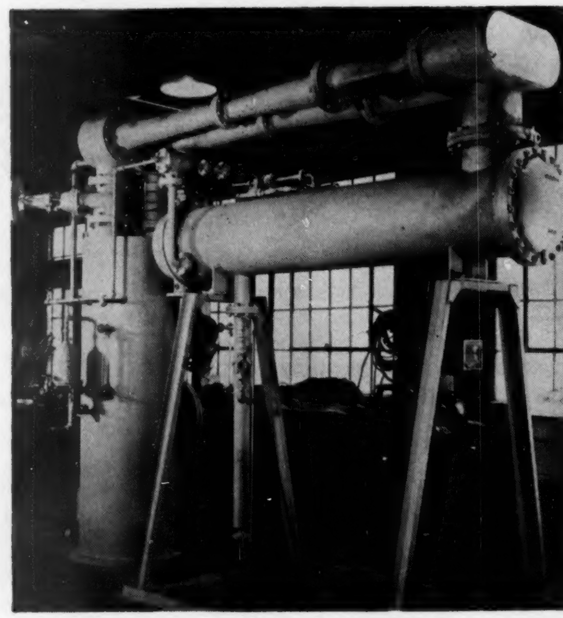
To understand more readily how this is accomplished, it would be well to review a standard steam vacuum refrigeration cycle and to analyze some characteristic performance curves. Briefly, the conventional cycle is as follows: Fig. 1 shows how the warm water to be chilled is admitted to the flash tank (1) at the connection (2) through which it flows to the distributors (3). The vacuum on the flash tank is produced by the booster steam ejector (4) that draws off all flashed vapors and discharges these vapors to the main condenser (5). Chilled water temperature is determined by the vacuum maintained in the flash tank by the booster ejector. A vacuum of 7.63 mm is equivalent to 45 F, a vacuum of 9.2 mm is equivalent to 50 F. Propelling steam, either high or low pressure (for example as low as 2 psig), is admitted to the booster at connection (6). A higher absolute pressure (as compared to the flash tank vacuum) is maintained in the main condenser by the two stage ejector with its intercondenser (7). This ejector may be operated at steam pressures of 35 psig or higher. Higher steam pressures are preferred, but when only low pressure steam is available this ejector system may be replaced by a mechanical vacuum pump.

The water thus chilled by the flashing of these vapors falls to the bottom of the flash tank and is removed by a hotwell type centrifugal pump (8). The cooling water for circulation to the main condenser is admitted at Point (9), as indicated. The level in the flash tank is maintained by the liquid level controller (10). From the foregoing, it is apparent that the system is extremely simple; that there

Elliot Spencer is Design Engineer with the Graham Mfg. Co. This paper was presented at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.



Completed assembly for shipment



Steam-vacuum system with surface-type condenser

are no moving parts other than the centrifugal chilled water pump.

For comparison purposes, we will examine the characteristics of the system in Fig. 1 under full load design conditions shown in the diagram. These conditions are given in Table I.

Such a system would use (at full load) 25 lb/hr of steam per ton of refrigeration and 6.6 gpm of water per ton of refrigeration. For this condition the condensing temperature would be 103 F.

Steam consumption and water requirements are given at full load and at the design wetbulb condition. The presumption is that water is coming from a cooling tower at 85 F at the design wetbulb condition. As the wetbulb drops, the water which comes from the cooling tower will be cooler and the steam consumption of the steam vacuum refrigeration system will be reduced.

Table I

Chilled Water Temperature	45 F
Condenser Water Temperature	85 F
Wetbulb Temperature	78 F
Drybulb Temperature	95 F
Steam Pressure	100 psig
Condenser Water Range	10 F

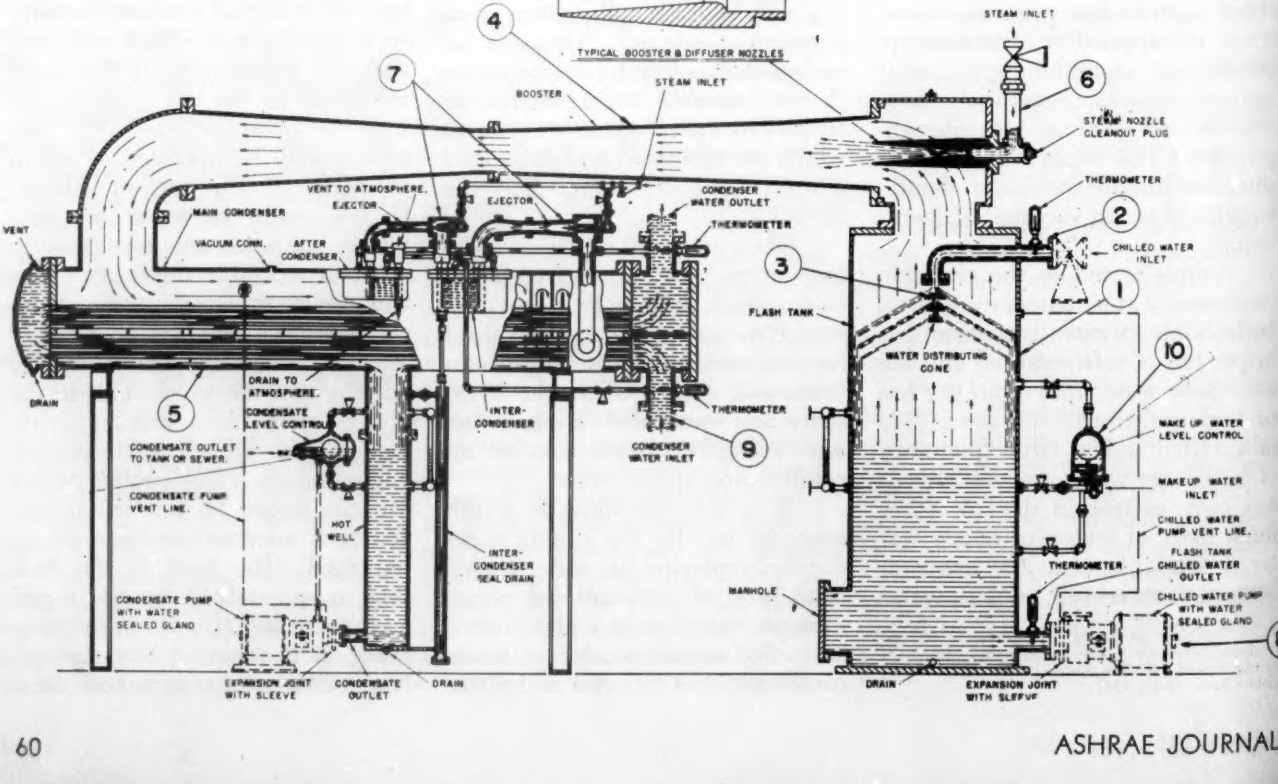


Fig. 1 Steam vacuum refrigeration system with surface type condenser. 1—Flash Tank. 2—Chilled Water Inlet. 3—Distributors. 4—Booster Steam Ejector. 5—Main Condenser. 6—Steam Inlet. 7—Intercondenser. 8—Centrifugal Pump. 9—Condenser Water Inlet. 10—Liquid Level Controller.

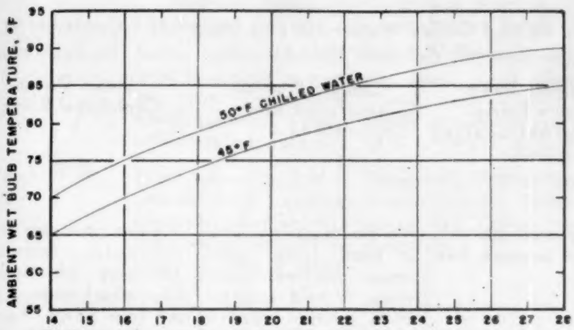


FIG. 2

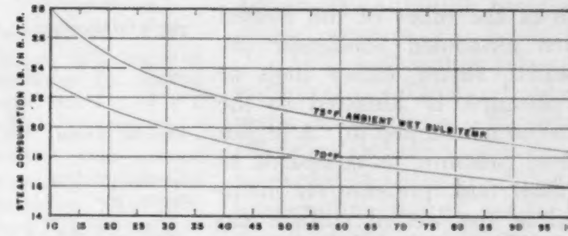


FIG. 3

Performance charts for steam-vacuum refrigeration systems vs. wet bulb temperatures

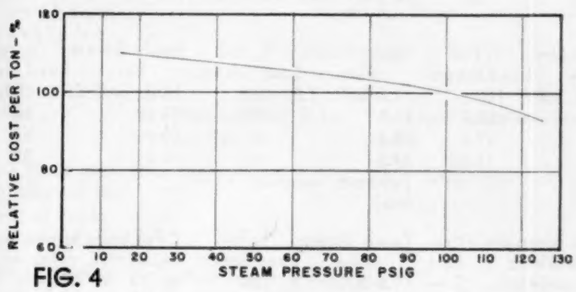


FIG. 4

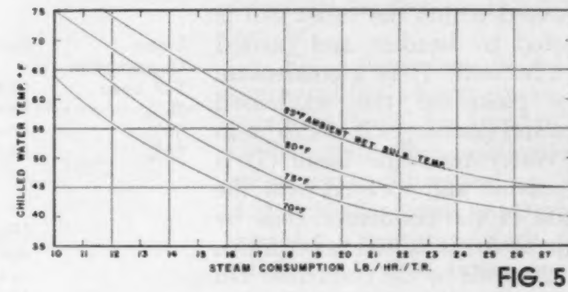
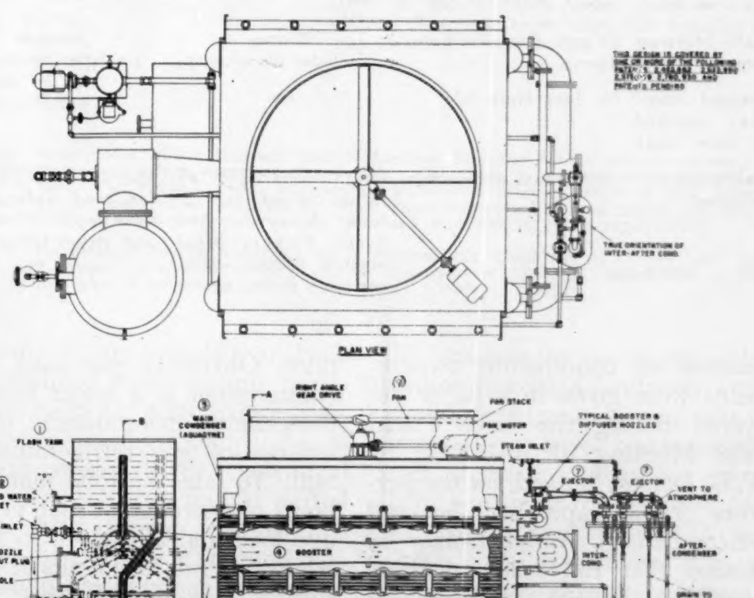


FIG. 5

Table II

	Lithium Bromide Absorption	Steam Driven Centrifugal Refrigeration
Steam	19.6 pph/T.R.	16 pph/T.R.
Condenser Water	3.5-4 gpm/T.R.	3 gpm/T.R.
Condenser Water Rise	16 F	20 F
Return Condenser Water Temp.	101 F	105 F

Fig. 1A Two-booster modified steam vacuum refrigeration unit with air-cooled wetted surface condenser



Comparable characteristics for lithium bromide absorption and steam driven centrifugals under full load design conditions would be similar to those shown in adjacent Table II.

In order to compete with other steam-operated refrigeration systems, steam vacuum refrigeration steam and water rates must be reduced.

Fig. 2 charts steam requirements of steam vacuum refrigeration systems vs wet bulb temperature. By reference to Fig. 2 we can see that if the condensing temperature is dropped to 90 F, the maximum full load design steam rate becomes 16 lb/hr per ton of refrigeration. This can be accomplished by replacing the cooling tower, condenser water pump and surface condenser, of Fig. 1 with an air-cooled wetted surface condenser.

Fig. 1A illustrates the revised steam vacuum refrigeration cycle which replaces the cooling tower and condenser of the conventional cycle with an air-cooled wetted surface condenser. In this system, the warm water to be chilled is admitted to the flash tank (1) at the connection (2) through which it flows to the distributors. The vacuum in the flash tank is produced by the booster steam ejector (4) that draws off all flashed vapor

and discharges this vapor to the inside of the tubes of the wetted surface air-cooled condenser (5). Propelling steam, either high or low pressure, is admitted to the booster at connection (6). A higher absolute pressure, as compared to the flash tank pressure, is maintained in the wetted surface air-cooled condenser by the two stage ejector with its inter and aftercondenser. The vapor from booster ejector (4) which enters the wetted surface air-cooled condenser (5) is condensed within the tubes and is collected by headers and carried into a hotwell. Here a condensate pump picks up the condensed vapor and discharges it to the basin (7). Water from the basin (7) is recirculated and sprayed over the outside of the condenser coils by pump (8). Air is caused to be drawn over the coils by the condenser fan (9). The recirculated water is thus cooled and, in turn, acts to cool the outside of the condenser tubes (10), condensing the vapors inside the tubes. A portion of the recirculated water is bypassed to the surface inter and aftercondensers of the two-stage air ejector system (11), wherein it is used to condense the motive steam and saturation vapors.

Vapors condensed in the intercondenser are piped to the suction of the main condensate pump; vapors condensed in the aftercondenser are drained to the basin. As can be seen by reference to the cycle shown in Fig. 1A, we have replaced the cooling tower and cooling tower pumps of Fig. 1 with the wetted surface condenser. By eliminating the temperature split between the cooling tower and the main condenser, we are able to make a closer approach to the prevailing atmospheric conditions, thus reducing the condensing temperature and consequently the steam consumption of the system. Reference to the characteristic curve of steam vacuum refrigeration systems, Fig. 2, indicates that any additional reduction of condensing temperature will result in a further saving in steam. Reduced condensing temperatures will result when outside climatic conditions are below design levels (even when full load is on the system), and also with reduced loads.

Average load requirements for seasonal air conditioning requirements were given in a paper delivered during the Lake Placid 1959 Meeting of ASHRAE by W. G. Dorsey, Jr., and, for the purposes of this paper, will be used for comparison. Author Dorsey indicated that full load operation occurs but 20% of the time, 75% load 30% of the time, 50% load 40% of the time, and 25% load 10% of the time. For year round systems, load requirements are often below 25% a large portion of the time. Further, during these periods, utilities selling steam often raise their price. Obviously, part-load steam consumption is a major factor in determining the economic characteristics of any refrigeration system. To take the total number of hours of operation at each percentage load and add them to obtain equivalent full-load hours of operation is inaccurate. Since steam vacuum refrigeration, centrifugal steam-driven refrigeration and lithium bromide absorption operate at varying efficiency at part-load levels, careful evaluation of loads and outside climate conditions and their relationships to the loads should

Table III—Comparison Chart of Various Steam Powered Refrigeration Systems

200 T. R. Systems Selected for 45 F Chilled Water—100 psig Steam—85 F Condenser Water —78-W.B. (see Note 6)

	Modified Steam Vacuum Refrig.	Lithium Bromide Absorption Refrig.	Steam Driven Centrifugal Refrig.
External cond. water—gpm	No external cooling water required.	3.5 to 4	3
Cond. water temp. rise	———	16 F	20 F
Effect of reduction in cond. temp. below design conds.	5 F drop produces 10% reduction in steam flow.	No steam saving. Drop in cond. temp. unbalances machine. Cond. temp. is held constant. With solution control steam need is reduced at part load. (See below).	Vacuum on steam cond. increases, increasing efficiency of turbine. Max effect approx. 7%. At low cond. temp. operational difficulties result.
Steam Consumption	Equiv. Steam Rate / % Full Load Steam	Equiv. Steam Rate / % Full Load Steam	Equiv. Steam Rate / % Full Load Steam
Full Load	16 lb/hr/T.R. / 100	19.5 lb/hr/T.R. / 100	16 lb/hr/T.R. / 100*
75%	14 / 65.7	20.8 / 80	14.45 / 68
50%	12 / 37.6	23.4 / 60	13.75 / 43
25%	10 / 15.60	34.3 / 44 (without solution control)	19.2 / 50
	(Figures given are at reduced wetbulb at part load conditions. See Note 6.)	Equiv. Steam Rate / % Full Load Steam 19.5 lb/hr/T.R. / 100 18.5 / 80 17.7 / 60 17.4 / 40 16.8 / 10 (with solution control)	* For units having larger capacities, rates as low as 13 lb/T.R. are possible.
Cond. Water Pump hp	2-hp recirculating spray pump on evaporative condenser.	40	20
Auxiliary hp	8½ approx. 1 lb/hr of steam will be used for each T.R. capacity for secondary ejectors.	10	5/6 (intermittent for purge unit) approx. 0.75 lb/hr of steam will be used for each T.R. for ejector on steam cond.
Cooling Tower or Mod. Steam Vac. Fan hp	20 (Fan hp)	20	15
Water Make-up gpm	33 gpm (includes blowdown).	90 gpm (includes blowdown)	66 gpm (includes blowdown)
Licensed engineer required by local code	No (see Note A).	No	Yes
Maintenance Required	Maint. fans and pumps.	Clean cond. and absorber. Maint. fans on tower & cond. water pump. Special treatment to protect against corrosion during winter.	Maintain cooling tower fan, cond. water pumps, centrifugal compressor and steam turbine.

Table III — Continued

	Modified Steam Vacuum Refrig.	Lithium Bromide Absorption Refri.	Steam-Driven Centrifugal Refrig.
Space Required	Entire system uses approx. same area as cooling tower for conventional systems. Entire unit installed outdoors.	Approx. 10% more cooling tower area than steam driven centrifugals plus indoor space of 5 x 19 x 8 ft high.	Approx. 10% less than absorption for cooling tower plus 8 x 15 x 7 ft high indoor space.
Weight	Total system weight is greater than cooling tower alone but less than total weight of conventional systems & tower.	Tower 10% heavier than centrifugal. System about 25% lighter than centrifugal.	Tower 10% lighter than absorption. System about 25% heavier than absorption.
Noise & Vibration	Low	Low	Low
Enclosed engine room usually required	No	Yes	Yes
Cond. water required	No (make-up for Evap. Cond. only)	Yes (plus make-up for tower)	Yes (plus make-up for tower)
Cost % of piping external to tower or Aquamizer	(See Note B)	100% plus	100%
Cond. water pump required	No (small pump included in system cost)	Yes	Yes
Relative Cost	0	100 plus	100
Cooling tower required	No	Yes	Yes
Relative Cost	0	100 plus	100
Refrigerant	Water	Water (Lithium Bromide Absorbant)	Refrigerant
Control	Automatic 0-100%	Automatic 0-100%	Automatic start-up manual
Relative installed first cost includ. cooling tower, steam piping, cond. water pumps and cond. water piping. Starters on all motors included	Entire unit on roof-outdoors. 80%	100 plus 10%. Tower on roof. Refrig. unit in basement. 100%. Tower on roof. Refrig. unit on roof in engine room. (Cost of engine room not included.)	100. Tower on roof. Refrig. unit in basement. 95. Entire system on roof. Refrig. unit indoors. (Cost of engine room not included.)

Note A: Local Codes vary but usually require Licensed Engineer for halocarbon systems. For both absorption and steam vacuum refrigeration no engineer is required when low pressure steam is used. For steam vacuum refrigeration when high pressure steam is used, Licensed Engineer may or may not be required, but when steam is supplied by local utility and no high pressure boiler is maintained on premises, Licensed Engineer is not a requirement.

Note B: No external cooling water piping is required but when unit is remote from steam source and competing systems would normally be located in basement, added cost of steam piping to unit must be considered.

be made in any complete analysis. The modified steam vacuum refrigeration cycle has performance characteristics which are comparable to that of steam driven centrifugal systems at full load and at partial loads. This system requires less steam than lithium bromide systems, and has eliminated the need for an external cooling water source.

The effect of steam pressure on any steam vacuum refrigeration cycle can be seen by reference to the curve of Fig. 3. Steam consumption increases as steam pressure is decreased. Regardless of this situation, by use of the wetted surface air-cooled condenser, steam vacuum refrigeration can now be designed to have maximum full load steam consumptions of 20 lb/hr per ton of refrigeration, even with steam available at 15 psig. This is equal to, or better than, the steam consumptions to be expected from absorption or steam driven centrifugal systems. At reduced loads, of course, steam consumption will be reduced as indicated earlier. Steam vacuum refrigeration systems designed for lower pressure steam with wetted surface air-cooled condensers, will be adversely affected by reduced steam pressure.

All steam vacuum refrigeration systems respond to increased chilled water temperature with improved efficiency and the modified cycle is no exception. Higher chilled water temperature means not only reduced steam consumption at all load conditions, but also reduced first cost due to decreases in equipment size. At higher chilled water temperatures, the modified system becomes smaller and lighter in weight.

Fig. 5 shows how chilled water temperature affects steam consumption (see Notes 1 and 6). Since chilled water temperature affects steam consumption and equipment size, it is sometimes worthwhile to consider a two-stage flash system when the chilled water temperature rise is higher than the conventional 10 F, for example 20 F as used on high rise coils. In such systems the returning chilled water is first cooled from say 65 to 55 F in one flash chamber, and then from 55 to 45 F in the second flash chamber. Approximately half of the refrigeration load is chilled at the 55 F level. Reference to Fig. 5 indicates that the lb of steam per hr per ton of refrigeration required for this portion of the load is less than that required for that portion of the load done at the 45 F level. The overall steam consumption will be lower than if all of the work were done at the final flash temperature of 45 F. Generally speaking, the response of steam vacuum refrigeration systems to higher chilled water temperatures is greater, proportionately, than the response of either absorption or centrifugal refrigeration systems.

Steam vacuum refrigeration systems may be hand-controlled, partially automatic, or fully automatic. The simplest way to regulate the capacity of steam vacuum refrigeration systems is to install several ejectors in parallel and operate only as many ejectors as are required to handle the heat load. A simple automatic on-off type of control is used for this purpose. By sensing the chilled water temperature out of the flash tank, the controller can turn the steam on or off to each ejector in con-

secutive order to vary the refrigeration capacity of the system.

When multiple boosters are used, the flash tank is divided into compartments (see Note 2); each compartment operating in conjunction with its own booster. As the load changes, either in the nature of an increase or a decrease, a booster will be turned on or off and the water valve of that compartment will be opened or closed to synchronize with the corresponding steam valve. This may be done either through the use of automatic controls or by hand. Each compartment is portioned off by a double baffle that extends to within about 12 in. of the bottom of the flash tank. This opening at the bottom, joining all compartments together, always is submerged.

When a booster is shut off, the compartment it services is then subject to a rise in pressure which causes the water level to depress, whereas the water level rises in the compartments that are under the higher vacuum of the operative booster. The flash tank is sufficiently deep to allow for these variations in the water level and still maintain a seal between compartments (see Note 2). Make-up water is admitted through a float controller that is regulated by the level in the compartments that are in operation. Failure of controls cannot damage the steam vacuum refrigeration unit, nor can faulty operation of the control system result in damage. Damage from freeze-up is not possible.

Table III is a comparison chart of the various steam powered refrigeration systems. Figures given are based upon the design rates shown therein. Although the modified steam vacuum system selected for this comparison shows a steam rate and first cost which favor it over both steam driven centrifugal and absorption systems, the steam vacuum system could be designed for a higher steam rate at a corresponding reduction in first cost. Under these conditions, the steam vacuum system would still be competitive in steam consumption with absorption systems. Thus, when only low pressure steam is available, a steam vacuum system may be selected for a peak rate slightly higher than the design peak rate for absorption. Part load rates, taking reduced wetbulb into consideration, are still lower than absorption (see Note 3), which leaves the steam vacuum system still at an advantage insofar as steam consumption is concerned.

The response of the modified steam vacuum system to lower condensing temperatures is a most important characteristic. A recent investigation of published weather data for a northern city indicated that out of 976 hr of an operating season, but 3 hr averaged above the wetbulb design condition, and but 54 hr were above the 70 F wetbulb condition; 440 hr were below 66 F wetbulb. The prevalence of wetbulb conditions at or above the design point will be influenced by the location of the specific installation. However, the proportion of time during which a refrigeration system operates at or near design conditions, insofar as wetbulb is concerned, is quite small, and any system which benefits substantially when these conditions are reduced offers a most important advantage. The resultant savings in steam consumption at lower wetbulb conditions make the modified steam vacuum system most attractive from an operating economy standpoint.

With no moving parts and only water as the refrigerant, maintenance and repairs are quite low. Records for even the earliest steam vacuum refrigeration systems indicate that repair bills averaged $20 per year. Life expectancy has been excellent, far greater than that of machines with moving parts. Maintenance experience with the new steam vacuum system has been too limited to be predicted accurately. However, since the low side of the system is no different from previous steam vacum refrigeration systems, it would be expected that maintenance experience on this part of the system would duplicate that on previous systems.

It is anticipated that maintenance on the high side (condenser) of this system would be approximately equivalent to that on a cooling tower. Maintenance of fans and recirculating water pumps is required for both. The recirculating water pump of the modified steam vacuum system would be smaller than the main condenser water pump of the cooling tower; however, the addition of the small condensate pump on the modified steam vacuum system would make up for this size differential. The overall maintenance requirements for the modified steam vacuum system will be less than the requirements for any other system.

One special advantage of the modified steam vacuum system is that the "fill" in the condenser is non-flammable, which means there is no off-season fire hazard such as there could be for wood fill cooling towers. The absence of moving parts on the low side of the system means there will be no vibration at either the high or low frequency level. Noise levels are also low, less than that of a fan and water on a cooling tower. There is no "hunting" noise such as can occur on other systems operating at extreme low loads.

The cost of installing any system is hard to evaluate, depending upon where the unit is installed, whether it is located on the roof or in the basement. The steam vacuum system usually is installed outdoors; in the case of a tall building it is installed on the roof where the cooling tower usually would be installed. The rigging cost of the modified steam vacuum system is usually less than, or about equal to, the rigging cost of a cooling tower alone. If it is planned to install the entire refrigeration system on the roof or in a penthouse, then rigging costs of the other systems would differ little from the rigging costs of a modified steam vacuum system. When it is planned to install the other refrigeration systems in the basement and the cooling tower on the roof, the combined rigging costs of the two procedures should be larger than that for the modified steam vacuum system installed on the roof. Since the steam vacuum system customarily is installed out of doors, it would require no engine room. In addition to the saving brought about by elimination of the engine room, rigging should be simplified due to the elimination of encumbrances produced by walls, etc. The cost of condenser water piping becomes a major factor when comparison is made between a modified steam vacuum system installed on the roof and any other system in-

stalled in the basement with a cooling tower on the roof. The cost of steam piping will be increased since a steam line must be run from the street to the roof.

Table III provides relative costs of the various refrigeration systems. Investigation has shown that the overall first cost of the modified steam vacuum system is lower than that for other systems, with the possible exception of electric-driven centrifugal systems when these systems are installed in penthouses immediately adjacent to the cooling tower. In arriving at these evaluations, the elimination of protective housing for the steam vacuum system has not been taken into consideration. This saving, especially in existent buildings, will be quite advantageous to the steam vacuum system. At higher chilled water temperatures, the modified steam vacuum system will prove even more attractive than at the temperatures considered.

The modified steam vacuum system occupies approximately the same space normally required for the cooling tower alone for conventional steam-operated systems. Dimensions for a typical 225-ton, 45 F, modified steam vacuum system approximate those for a 300-ton, 50 F chilled water system. Moreover, space requirements for larger tonnages would not necessarily increase proportionately. Since the entire system is installed where the cooling tower alone would have been installed, machine room space is saved for other uses.

Weight requirements of the modified steam vacuum system are greater than the weight of the cooling tower alone for steam-operated refrigeration systems, but the weight of the entire system is about equal to, or slightly less than, the combined weight of a cooling tower and steam-operated refrigeration system where the entire system is installed on the roof. Where it is required to locate the modified steam vacum system on a lower floor set-back, the system would have the disadvantage of losing some of the available chilled water pumping head. This occurs when the returning chilled water is flashed to the vacuum in the evaporator. Closed chilled water cycles available in other systems do not have this disadvantage. In both lithium bromide absorption and centrifugal refrigeration, water is chilled under the operating static head pressure, but in the modified steam vacuum system the water must be flashed to the vacuum in the evaporator, resulting in a static head loss and consequent greater pumping horsepower. This is not the condition when the entire system is located on the roof.

Although the modified steam vacuum system operates under a vacuum (see Note 4), this vacuum exists only up to the discharge side of the chilled water pump. Water returning to the evaporator is under a positive pressure. This means that the only air leaks that can occur must take place within the system itself. Proper design and minimization of joints through which leakage can occur have made air leakage a negligible problem in these systems.

As a further precaution, vacuum is maintained in these systems by a large two-stage ejector system with air-handling capacities much greater than those found on either absorption or centrifugal systems. The danger of air leakage in steam vacuum refrigeration systems is only slightly greater than the danger of air leakage in the surface condenser and turbine combination of a centrifugal refrigeration system. Experience has indicated that this problem is practically non-existent, except during the original start-up of the system (see Note 5).

In summary, a new steam-operated refrigeration system is now available for comfort air conditioning; a system with low installed first cost, low annual steam consumption, and one which is reliable and maintenance-free. Although limited in size at the present time to systems under 500 ton of refrigeration in a single package, the merits of this modified steam vacuum system should be considered wherever steam is used to produce refrigeration for air conditioning or other water chilling purposes.

NOTES

Note 1: Characteristic curves (Fig. 5) are for a series of boosters, each designed for the specific conditions shown on the curve.

Note 2: Description of part-load control is applicable to the systems shown in Figs. 1 and 1A. The modified steam vacuum system and the vertical flash tank systems shown in the illustration use a stacked flash tank eliminating the need for shutting water valves as boosters are turned on and off and also, incidentally, eliminating any inefficiency which would result from water stored in the now operating flash tank.

Note 3: Absorption refrigeration part-load rates have recently been improved by a technique known as solution control. An accurate representation of part-load performance of any given ejector system cannot be obtained by reference to Fig. 5, which represents the performance of a series of boosters; each designed for the specific conditions shown on the curves. A characteristic throttling curve would more accurately depict performance for any system selected. Thus, performance of a system selected for operation at 90 F condensing temperature would not be as good as shown in Fig. 5 when this system operated at 80 F condensing temperature; i.e. the throttling curve would not be a straight line. The figures given, however, are accurate enough for the purpose of this article.

Note 4: Early steam vacuum refrigeration systems were troubled by loss of vacuum due to poor piping techniques and insufficient ejector capacity. Modern systems have ejectors with capacities many times those of earlier systems and several times the capacity of the steam ejectors used on the surface condensers of all steam-driven centrifugal systems. By designing the system so all piping is under a positive pressure, leakage problems can be limited to the steam vacuum refrigeration unit itself. Present day steam vacuum refrigeration systems encounter no difficulty with maintenance of vacuum.

Note 5: Breaking.
Loss of vacuum and insufficient condenser capacity also were responsible in early systems for an effect known as breaking.
When a booster ejector breaks, the motive steam discharges to the flash tank resulting in a heating rather than a cooling process. Breaking occurs when the compression range over which the ejector is required to function exceeds that for which it was designed. This can occur if the condensing temperature is higher than design.

Excessive condensing temperature can be eliminated by simply designing the evaporative condenser portion of the unit with sufficient excess surface to anticipate the worst conditions found in local weather records for the installation in question.

An over-riding control device can be incorporated in the system which will permit the chilled water temperature to rise a few degrees during those few hours when outside W.B. conditions exceed the maximum which was anticipated during design. This rise in chilled water temperature will decrease the range over which the ejector must compress and eliminates any danger of breaking.

Proper design of condenser and ejector with a 15% safety factor in condensing surface and a booster designed to operate without breaking at condensing temperatures 2-3 F above design will not only insure against breaking but will improve performance at all load conditions since excess surface will result in reduced condensing temperature at all loads.
Breaking can only occur at full load when condensing conditions exceed design, or if load is greater than design. This can be accommodated by the steps described above at all other times; i.e., when load is less than design or when W.B. is less than design, there is no possible danger of breaking.
Breaking can only occur on a poorly designed system.

Note 6: As load is reduced, condensing temperatures are reduced, since at reduced load evaporative condenser is oversized and closer approach to wetbulb occurs—at 100% W.B. has assumed 78 F, at 75% load W.B. is 73 F, at 50% of load W.B. is 70 F, at 25% load W.B. is 64.5 F. It must be emphasized that any comparison of the relative energy requirements of one system, as opposed to another, must be made by a detailed analysis of the variation of load with wetbulb. Load does not necessarily drop off as the wetbulb is reduced. Internal load proportions, fresh air requirements, etc., will make a great difference. Analysis must be made for each specific case. A 25% load at 78 W.B. would require more steam per T.R. than a 25% load at 70 W.B. The above figures, however, do reflect the dramatic reduction in steam requirements as load and W.B. decrease.

PRESIDENT'S PAGE

A Visit Overseas

JOHN EVERETTS, JR.
President
ASHRAE

Paris is a beautiful city, vivacious, lively, feminine. London has a stoical charm steeped with solidarity, heraldry and protocol. These were our impressions as Mrs. Everetts and I visited these great cities this past September. Each city is in itself a paradox as far as the heating, refrigerating and air conditioning industry is concerned, yet they are vitally important to us.

Approximately only 10% of the residents have central heat and mechanical refrigeration in Paris. It was interesting to see the housewives in the morning do their marketing, which they have to do every day because of lack of refrigeration, traveling on their motor bikes, bicycles, or walking to the meat market, grocers, fruit and vegetable market, and bakery to buy their daily requirements of food. There are no supermarkets, as we know them. Each of the above stores is separate and independent and only the meat market showed any indication of even limited refrigeration in its display cases. Very little meat was kept in the store overnight.

Heating, except in the more modern hotels and office buildings, is done with fireplaces. In spite of this seeming backwardness in heating and refrigeration, Paris is the headquarters of the International Institute of Refrigeration, which Dr. R. C. Jordan so aptly described in the August issue of the JOURNAL in the President's Page. I visited these headquarters and held some valuable conferences to the mutual benefit of both our Societies.

The IIR was having its International meeting in Cambridge, England, and it was there that I was able to spend some time with Dr. Thevenot, Director of the Institute, discussing further some of our mutual problems. There is no question that the IIR is the international leader as a technical society in the field of refrigeration, and they recognize ASHRAE as the international leaders in heating and air conditioning. Their problems are quite comparable to ours. Food preservation, cryogenics, refrigerated storage, heat transfer, and a myriad of others.

The big problem, however, is the lack of communications between our Society and IIR, which is the world center of the science of refrigeration. After visiting with many members of IIR from all over the world, I cannot help but emphasize Dr. Jordan's statement in his article, when he said, "Any egocentric beliefs in our own universal technical superiority have been shattered by the achievements of the Russians, the Red Chinese and others. We have wakened to the fact that many of the fundamental technical contributions have been originated beyond our own shores and that in the quickening pace of our present civilization we must participate in the world-wide community of scholars and engineers or lag behind."

I believe our conferences were fruitful in the many suggestions which were offered as to how we all could cooperate in bringing their news and views to our attention, and ours to theirs. We cannot and must not lose sight of the fact that we are in strong competition, technically and scientifically with many other parts of the world and if we wish to maintain our leadership in the international scientific world then we must join forces and open our communications in a spirit of universal cooperation.

The suggestions which I received from IIR and also recommendations on the part of our International Relations Committee will be passed on to the Publications, International Relations, and Public Relations Committees for their evaluation and recommendations.

London, a city of over eight million population, also only has 10% of its residents heated with central heating systems and slightly over 10% of them have mechanical refrigeration. Air conditioning is confined primarily to industrial processes. Except for two office buildings now under construction, there is practically no commercial air conditioning. Commercial refrigeration is limited to storage warehouses and a limited amount of commercial food storage. High temperature hot water heating and panel heating in commercial structures is well advanced.

Plastics piping, plastics casing for air washers, and plastics fan wheels are being widely used and promoted in commercial buildings for small air handling equipment, plumbing drains, vents and wastes. In Paris, we saw flexible plastics piping being used for underground gas and water mains.

The main reason for our visit to London was to attend the Institution of Heating and Ventilating Engineers Conference and the First International Exhibition of Heating, Ventilating, Air Conditioning and Refrigerating Equipment.

The Conference consisted of 39 technical papers covering research, education, heating and ventilating, air conditioning, hospital practice, heat transfer, measurements, and other allied subjects. Two papers were presented by U. S. members of ASHRAE, Prof. Burgess Jennings and Prof. A. C. Min. The Conference sessions were attended by 800-900 delegates representing 24 countries throughout the world. Interpreters were provided for German and French translations.

Dr. F. M. H. Taylor, President of I.H.V.E., in his opening address stressed the importance of coordinated research, long range planning and greater attention to education. He, as well as a number of the speakers, brought out the lack of communications throughout the world which was costing time and money in the duplication of research as well as development. Dr. Taylor also recommended that a common center be established for the exchange of information and research papers. This is now being done, of course, through the IIR Bulletins for refrigeration and some air conditioning. There is no central agency for heating and ventilating.

The subject of communications was stressed in conversations with delegates from Germany, Holland, Sweden and France. It is time, I believe, that this problem be recognized for its importance and action taken to meet the situation for the benefit of our Society and other societies in our allied field.

The exhibition was considered a great success. Approximately 220 exhibitors used 7,500 sq ft to show their products, some of which came from the U. S., Germany, Sweden and other foreign countries. The thing that impressed me most was the way the exhibits were "dressed up" with flower boxes placed here and there and filled with flowering plants of many colors. This made the exhibits look "alive" and not like a stock pile of machinery parts in an inventory bin.

Altogether, I considered the trip well worth while, particularly the personal contacts with people from many countries who have interests and problems common to our own.

An instrument for the *Measurement of the Humidity* of air

J. D. WENTZEL

The problem of accurate humidity measurement is one which concerns not only meteorologists, but also many manufacturing industries. It is often necessary to control the amount of water vapor in the air artificially to comply with specific requirements. The accurate continuous measurement of the humidity of the air is, therefore, of the utmost importance.

At present there are various types of humidity-measuring instruments in use but none of these fully complies with all the requirements necessary in practical applications. This paper relates to the characteristics of a few existing instruments commonly used and also describes the development of a new instrument based on the principle of the adiabatic saturation of a sample of air. The same principle is utilized as in the wet- and dry-bulb hygrometer, but the new instrument eliminates some of the inherent disadvantages of the latter type of hygrometer.

Mechanical humidity instruments, for example the hair hygrometer,[1] are extremely simple to operate but lack accuracy. On the other hand, dew-point measuring instruments,[2] designed for continuous recording, are complicated and expensive. Electrical instruments,[3] using a hygroscopic salt as sensing element, are sensitive to small changes in the moisture content of an air sample, but the calibration of these instruments appears to be unstable. In Table I a summary is given of the main features of the most commonly used humidity measuring instruments.

Conventional wet- and dry-bulb psychrometer – The wet- and dry-bulb psychrometer, which yields a high degree of accuracy when used carefully,[4,5,6,7,8,9,10,13] is used quite commonly in industry. To evaluate the absolute humidity of an air sample by means of wet- and dry-bulb measurements, the following psychrometric equation[4,5,7,8,9,12,14] generally is used:

$$e = e'_s - AP\,(\theta - \theta') \qquad (1)$$

where

e = partial vapor pressure of the air sample
e'_s = saturated partial vapor pressure at wet-bulb temperature
θ = dry-bulb temperature
θ' = wet-bulb temperature
P = total atmospheric pressure
A = psychrometric constant

To evaluate the psychrometric constant A, many theories[4,5,6,7,8,11,12,13,14,15] have been advanced, of which August's convection theory[4,5,16] is perhaps the most commonly known.

August assumed that a stagnant film of air surrounds the wet-bulb element, and this film of air is saturated continuously with water vapor; the latent heat necessary to evaporate the water is supplied by the air stream flowing over the wet-bulb.

August's theory holds good when the speed of ventilation over an infinitely small wet element is quite high, but fails to explain what happens at the intermediate ventilation rates and when relatively large thermo-elements are used, as in actual practice.

Carrier[6,13] worked on the same assumptions and further assumed that any make-up water necessary to keep the evaporating surfaces wet is supplied at this ideal equilibrium temperature of the wet-bulb. He defined this equilibrium temperature as the temperature of adiabatic saturation and derived the following equation:

$$w = w'_s - \frac{C_{ph}}{L_1}(\theta - \theta') \qquad (2)$$

where

w = specific humidity of the air sample
w'_s = saturation specific humidity at the temperature of adiabatic saturation
C_{ph} = specific heat of the humid air sample
θ = dry-bulb temperature
θ' = temperature of adiabatic saturation (ideal equilibrium temperature)
L_1 = latent heat of evaporation at the adiabatic saturation temperature
$\frac{C_{ph}}{L_1}$ = psychrometric constant

Equation (2) can be rewritten[14] in the same form as Equation (1), i.e.

$$e = e'_s - AP\,(\theta - \theta')$$

where

$$A = \left[(1 - \frac{e'_s}{P})(1 - \frac{e}{P}) \cdot (\frac{\rho a}{\rho v} \cdot \frac{Cpa}{L_1})\right] \times \left[1 + \frac{e}{P - e} \cdot \frac{\rho v}{\rho a} \cdot \frac{Cpv}{Cpa}\right] = \left[\frac{\rho a}{\rho v} \cdot \frac{Cpa}{L_1} \cdot (1 - \frac{e'_s}{P})\right] \times \left[1 + \frac{e}{P}(\frac{\rho v}{\rho a} \cdot \frac{Cpv}{Cpa} - 1)\right] \qquad (3)$$

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Table I. Characteristics of a Few Humidity Measuring Techniques

	Gravimetric	Dew Point	Mechanical	Electrical	Wet- and Dry-bulb Thermometers	Wet- and Dry-bulb Thermocouples
Principle	Absorption by Chemicals	Condensation on a Mirror	Variation in Dimensions Due to Moisture Absorption	Variation in Electrical Resistance	Evaporation from a Wetted Surface	Evaporation from a Wetted Surface
Dew Point Range	—100 F→160 F	—100 F→120 F	15 F→150 F	—100 F→150 F	7 F→160 F	7 F→160 F
Accuracy	2% of Absolute Humidity	± 5 F at —90 F ± 2 F at —20 F ± 0.5 F at 70 F	± 5% Relative Humidity	± 1.5 Relative Humidity	± 1 F Difference Between Wet- and Dry-bulb	± 0.2 F Difference Between Wet- and Dry-bulb
Error at 70 F and 50% relative humidity	0.8%	± 1.8%	± 5%	± 1.2%	± 3.6%	± 1.0%
Dry-bulb Range	Unlimited	—100→120 F	32 F→150 F	—50→150 F	32 F→160 F	32 F→160 F
Reaction Time	Few hr	15 Sec to Several Min	5 Min	1 Sec to Several Min	At Least Two Min	15 Sec
Simplicity of Operation	Very Bad	Moderate	Very Good	Moderate	Good	Good

where

e = partial vapor pressure of the water vapor in the air sample

e'_s = partial pressure of the saturated vapor at the adiabatic saturation temperature Θ'

P = atmospheric pressure

Θ = dry-bulb temperature

Θ' = adiabatic saturation temperature

$\frac{\rho v}{\rho a}$ = ratio of the densities of the water vapor and dry air at the same pressure e'_s and temperature Θ'

Cpa = specific heat of air at constant pressure.

Cpv = specific heat of water vapor at constant pressure

L_1 = latent heat of evaporation at the adiabatic saturation temperature Θ'

Using Equation (2) in his calculations, Carrier presented a psychrometric diagram in 1911, which has since been in common use.

Various psychrometric equations have been deduced by different research workers to evaluate partial vapor pressure of a vapor-air mixture from wet- and dry-bulb readings, but the form of the various psychrometric equations is the same as in Equation (1). However, the proposed value of the psychrometric constant varies[4,5,6,7,8,9,11,12,13,15,16] as shown in Table II. The influence of the variation in the value of the psychrometric constant on the calculated vapor pressure is shown in Table III. It is clear that the calculated vapor pressure variation obtained by using the different values of the psychrometric constant, A, is greater for low than for high humidities.

Table II. Values of the Psychrometric Constant

θ	θ'	(θ-θ')	1 Adiabatic Saturation × 10⁻¹	2 Muller-Cosna × 10⁻¹	3 August × 10⁻¹	4 Whipple × 10⁻¹	5 Ferrel × 10⁻¹	6 Lambrechts × 10⁻¹	7 Carrier × 10⁻¹	8 Grant × 10⁻¹	9 Barenbrug × 10⁻¹	10 Sprung × 10⁻¹
120	120	0	3.314	3.458	3.750	3.243	3.876	3.761	3.271			
	119	1	3.323	3.456	3.748	3.256	3.873	3.759	3.284			
	118	2	3.334	3.455	3.746	3.267	3.871	3.757	3.296			
	116	4	3.353	3.451	3.742	3.290	3.866	3.753	3.319			
	112	8	3.388	3.443	3.733	3.331	3.857	3.745	3.365			
	110	10	3.402	3.439	3.729	3.349	3.852	3.740	3.381			
	100	20	3.466	3.420	3.709	3.428	3.829	3.720	3.461			
100	100	0	3.472	3.392	3.712	3.431	3.829	3.720	3.461			
	99	1	3.478	3.391	3.710	3.437	3.826	3.718	3.468			
	98	2	3.484	3.389	3.708	3.445	3.824	3.716	3.476			
	96	4	3.491	3.385	3.704	3.455	3.819	3.712	3.487			
	92	8	3.507	3.378	3.696	3.477	3.810	3.704	3.510			
	90	10	3.515	3.374	3.692	3.487	3.805	3.699	3.520			
	80	20	3.542	3.356	3.672	3.523	3.782	3.679	3.559			
80	80	0	3.546	3.325	3.672	3.523	3.782	3.679	3.559			
	79	1	3.547	3.324	3.670	3.526	3.780	3.677	3.562			
	78	2	3.550	3.322	3.668	3.529	3.778	3.675	3.565	3.67	3.613	3.677
	76	4	3.553	3.318	3.664	3.534	3.773	3.671	3.571			
	72	8	3.558	3.311	3.656	3.543	3.764	3.662	3.580			
	70	10	3.560	3.307	3.652	3.546	3.759	3.658	3.584			
	60	20	3.569	3.290	3.633	3.559	3.735	3.638	3.598			
60	60	0	3.570	3.262	3.634	3.560	3.735	3.638	3.599			
	59	1	3.570	3.260	3.632	3.560	3.733	3.635	3.600			
	58	2	3.572	3.258	3.630	3.561	3.731	3.633	3.600			
	56	4	3.571	3.254	3.626	3.562	3.726	3.629	3.602			
	52	8	3.569	3.247	3.618	3.563	3.717	3.621	3.604			
	50	10	3.567	3.244	3.614	3.563	3.712	3.617	3.604			
	40	20	3.562	3.227	3.595	3.560	3.689	3.596	3.604			
40	40	0	3.565	3.193	3.594	3.559	3.689	3.596	3.604			
	39	1	3.564	3.191	3.592	3.559	3.686	3.594	3.603			
	38	2	3.562	3.189	3.590	3.558	3.685	3.592	3.603			
	36	4	3.560	3.186	3.586	3.556	3.679	3.588	3.602			
	32	8	3.555	3.180	3.579	3.554	3.670	3.580	3.600			
			T	T	T	T	E	E	E	E	T	T

Θ = Dry-bulb temperature F
Θ' = Wet-bulb temperature F
(Θ-Θ') = Wet-bulb depression F

T = T = Theoretical values
E = E = Empirical values

Attempts have been made to determine the value of the psychrometric constant experimentally or to find an empirical relation between the measured wet- and dry-bulb temperature and the prevailing humidity. This has been done by Ferrel,[11,13] Carrier,[6,12] Grant[9] and Lambrecht. Unfortu-

Table III. Vapor Pressures Calculated From the Psychrometric Equation Using Different Psychrometric Constants

		1	2	3	4	5	6	7
θ	θ-θ'	e	e	e	e	e	e	e
120	1	3.3427	3.3418	3.3413	3.3416	3.3428	3.3420	3.3418
	2	3.2419	3.2402	3.2392	3.2397	3.2421	3.2405	3.2401
	4	3.0464	3.0432	3.0412	3.0423	3.0466	3.0437	3.0431
	8	2.6803	2.6745	2.6707	2.6730	2.6806	2.6757	2.6744
	10	2.5087	2.5019	2.4973	2.5001	2.5095	2.5034	2.5017
	20	1.7557	1.7453	1.7372	1.7482	1.7565	1.7482	1.7450
100	1	1.8662	1.8657	1.8653	1.8656	1.8662	1.8659	1.8657
	2	1.8014	1.8005	1.7997	1.8002	1.8015	1.8008	1.8004
	4	1.6761	1.6743	1.6728	1.6739	1.6761	1.6748	1.6741
	8	1.4416	1.4382	1.4353	1.4376	1.4415	1.4392	1.4381
	10	1.3319	1.3279	1.3244	1.3272	1.3418	1.3294	1.3277
	20	0.8515	0.8449	0.8392	0.8445	0.8505	0.8478	0.8446
80	1	0.9899	0.9895	0.9892	0.9896	0.9898	0.9897	0.9895
	2	0.9485	0.9479	0.9474	0.9478	0.9484	0.9482	0.9478
	4	0.8684	0.8672	0.8662	0.8672	0.8682	0.8677	0.8671
	8	0.7186	0.7163	0.7144	0.7165	0.7182	0.7175	0.7162
	10	0.6484	0.6456	0.6433	0.6459	0.6478	0.6471	0.6454
	20	0.3398	0.3346	0.3313	0.3363	0.3382	0.3375	0.3343
60	1	0.4944	0.4941	0.4939	0.4943	0.4943	0.4943	0.4941
	2	0.4676	0.4671	0.4668	0.4672	0.4674	0.4674	0.4670
	4	0.4156	0.4146	0.4140	0.4150	0.4153	0.4151	0.4345
	8	0.3178	0.3157	0.3147	0.3168	0.3171	0.3169	0.3156
	10	0.2716	0.2690	0.2679	0.2704	0.2707	0.2705	0.2688
	20	0.0662	0.0606	0.0596	0.0640	0.0625	0.0625	0.0603
40	1	0.2292	0.2289	0.2289	0.2291	0.2291	0.2291	0.2289
	2	0.2110	0.2105	0.2104	0.2108	0.2108	0.2108	0.2104
	4	0.1755	0.1744	0.1743	0.1752	0.1750	0.1749	0.1743
	8	0.1077	0.1054	0.1054	0.1073	0.1069	0.1066	0.1053

1 Adiabatic saturation
2 Grant
3 Ferrel
4 Lambrechts
5 Carrier
6 Barenbrug
7 Sprung

θ = Dry-bulb temperature F
θ-θ' = Wet-bulb depression F
e = Partial vapor pressure in in. Hg.

nately but little detail is known about the experimental procedure, dimensions of the thermometers and the specific types of wick used. It is, therefore, impossible to correlate the different test results.

August,[16] Whipple,[7] Muller-Cosna,[15] Barenbrug,[8] Sprung,[10] Arnold[4] investigated the problem theoretically. Differences in basic assumptions and different ways of resolving complicated equations into simpler forms, resulted in expressions which do not yield the same value of the psychrometric constant when used in calculations.

Any factor causing the depression of the wet-bulb to be incomplete will have to be compensated for by the use of a higher psychrometric constant. This explains the discrepancies between the constants shown in Table II.

If, however, the adiabatic saturation temperature is known, the applicable psychrometric constant may be calculated from Equation (3), thus eliminating an inherent weakness of the wet- and dry-bulb psychrometer.

The instrument now described has been designed to fulfill this purpose.

Description of the new instrument – The instrument designed to measure adiabatic saturation temperatures consists of a well insulated tube in which air and water are mixed adiabatically. Fig. 1 shows a diagrammatic layout.

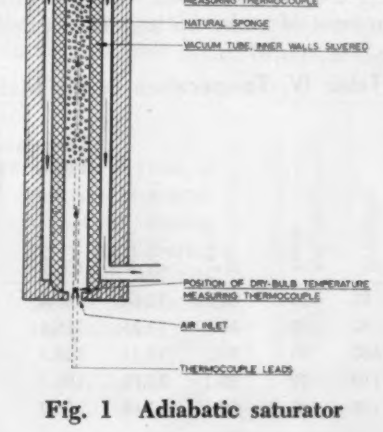


Fig. 1 Adiabatic saturator

To ensure adiabatic mixing of the water and the air, the mixing chamber consists of a vacuum insulated tube, the inner walls of which have been silvered to reduce heat flow. As a further precaution the mixing chamber was covered with plastics insulating material wrapped in aluminum foil. An air flow velocity of at least 50 fpm through the mixing chamber of the instrument was maintained by means of an air-ejector pump.

As the air flows through the sponge it becomes saturated and reaches an equilibrium temperature which is a function of the dry-bulb temperature and the humidity of the incoming air.

Make-up water may be supplied either intermittently or continuously, the effect of which will be discussed later. The equilibrium temperature of the air-water mixture is measured in the sponge near the end of the saturating chamber on the outlet side while the dry-bulb temperature of the air is measured in the vacuum tube near the entrance.

Theoretical consideration of the principle of operation – The new instrument is based on the principle of mixing a sample of air (of which the humidity is to be determined) with water in such a way that the process takes place adiabatically. The rate at which the make-up water is supplied relative to the rate of evaporation as well as the initial heat content of the sample of air and the make-up water determine the equilibrium temperature reached. The relative importance of the respective factors can be derived from a consideration of the heat balance of the system. Consider a unit weight of air flowing through a saturating chamber, Fig. 2. Air enters at a dry-bulb temperature t_1 F and a specific humidity w_1. Air leaves the chamber saturated at t_2 F and a specific humidity w_2. The surplus water supplied per unit weight of dry air is W, thus the total amount of water supplied at temperature t_3 F is $W + (w_2 - w_1)$, where W is the surplus water, which is supplied at temperature t_3 F and leaves the instrument at a temperature t_2 F.

The enthalpy of the water, air and water vapor is h_w, h_a and h_v,

respectively, at the temperature indicated by the subscripts.

For steady state conditions the heat balance may be written as follows:

$$W \cdot h_{wt3} + 1 \cdot h_{at1} + w_1 \cdot h_{vt1} + (w_2 - w_1) \cdot h_{wt3} = 1 \cdot h_{at2} + w_2 \cdot h_{vt2} + W \cdot h_{wt2}$$

$$h_{wt3} = \frac{ha\,(t_2 - t_1) + w_2 \cdot h_{vt2} - w_1 \cdot h_{vt1} + W\,(t_2 - t_3)}{w_2 - w_1}$$

For the condition when all the water supplied is evaporated, W = O, and

$$(t_3 - 32) = \frac{0.24\,(t_2 - t_1) + w_2\,h_{vt2} - w_1\,h_{vt1}}{w_2 - w_1} \quad (4)$$

where

$$h_{wt3} = 1 \cdot (t_3 - 32)$$

If the temperature of the make-up water, t_3, is higher than the equilibrium temperature t_2, some of the heat lost by evaporation has to be used to cool the make-up water. Thus, the equilibrium temperature will be higher than the value reached in the case where the water is supplied at a temperature equal to the true equilibrium temperature. This true equilibrium temperature is that temperature which will be reached near the exhaust of this instrument when the temperature of the make-up water is equal to the true equilibrium temperature. The true equilibrium temperature is, therefore, obtained only in cases where all the heat required for the evaporation of the water is surrendered by the air alone, and is equal to the adiabatic saturation temperature.

It is, theoretically, possible to supply the make-up water in the case of this new instrument at the adiabatic saturation temperature. In practice, however, it is more convenient to supply the water at the dry-bulb temperature. The measured equilibrium temperature will then differ from the adiabatic saturation temperature. This error will be increased when the rate of water supplied to the instrument is much larger than the rate of evaporation. Thus, if an error in the measured equilibrium temperature of 0.1 F is considered acceptable, Table IV shows the maximum temperatures at which the make-up water may be supplied for different rates of flow of the make-up water and for different rates of evaporation.

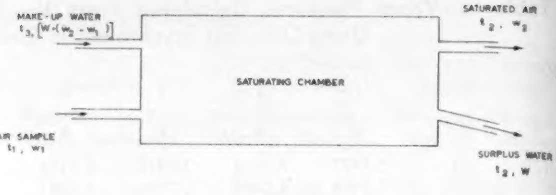


Fig. 2 Diagrammatic layout of adiabatic saturator

From this table it is clear that when the aim is to obtain an equilibrium temperature differing by a small amount from the adiabatic saturation temperature, the temperature difference between that at which the water is supplied and the adiabatic saturation temperature must be decreased as the ratio of the rate of water supply to the rate of evaporation increases.

When the make-up water is supplied at a rate equal to the rate of evaporation and at a temperature equal to the dry-bulb temperature, and when the difference between the equilibrium temperature and the adiabatic saturation temperature must not exceed 0.1 F, there is a limiting dry-bulb temperature for each adiabatic saturation temperature, as shown in Fig. 3. Below this limiting dry-bulb temperature the error will be smaller than 0.1 F, but above this temperature the error will be greater than 0.1 F.

The amount of water that has to be supplied to saturate an air sample at its corresponding adiabatic temperature when the dry-bulb temperature is at its limiting value is shown as a function of the equilibrium temperature in Fig. 4.

For conditions where the amount of make-up water supplied is much more than the rate of evaporation and is supplied at the dry-bulb temperature, appreciable errors may occur. Table V shows the errors in the measured equilibrium temperature as well as the resulting error in the calculated dew point temperature for a condition where the amount of water continuously supplied at dry-bulb temperature, per pound of air drawn through the instrument, is equal to five times the rate of evaporation.

It may be concluded that with the new instrument, as well as with any wet- and dry-bulb instrument, the temperature, as well as the rate at which the make-up water is supplied, must be controlled carefully when accurate humidity determinations are required continuously.

In the case where the make-up water is cooled to a temperature which is close to the adiabatic saturation temperature before entering the saturating chamber, the rate of flow of the make-up water is of secondary importance.

This can be obtained by using two interconnected vacuum insulated tubes, as shown in Fig. 5, the one acting as a cooling chamber into which a surplus of make-up water is fed and the other tube as the saturating chamber into which the cooled overflow water from the cooling chamber is fed; a continuous measuring apparatus is obtained.

Water is fed from an overhead tank into the cooling chamber, tube B, which also is filled with natural sponge. This water is cooled

Table IV. Temperature of the Make-up Water for Various Rates of Flow

Dry-bulb	Ad. Sat. Temp	Equilibrium Temp	Dew point Temp	Temperature of make-up water for different ratios of the excess water supplied to the amount evaporated. $x = \frac{W}{w_2 - w_1}$ x=0	x=1	x=2	x=3	x=4	x=5
85	80	80.1	78.66	181.5	130.8	118.2	105.4	100.4	97.0
90	80	80.1	77.26	136.6	108.4	98.9	94.2	91.4	88.5
100	80	80.1	74.31	107.1	93.6	89.1	86.8	85.5	84.6
110	80	80.1	71.05	101.1	90.6	87.1	85.4	84.3	83.6
120	80	80.1	67.48	95.0	88.1	85.4	84.1	83.3	82.7

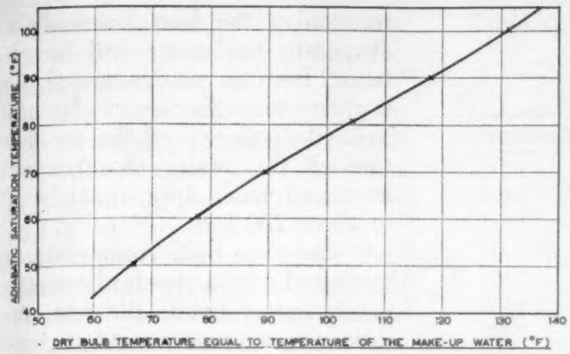


Fig. 3 Limiting dry-bulb temperature at which the make-up water must be supplied for selected equilibrium temperatures

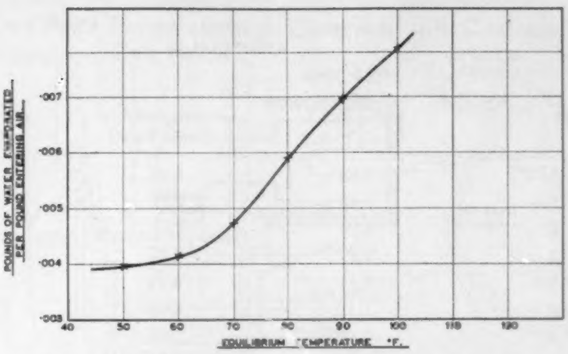


Fig. 4 Make-up water necessary to saturate an air sample at its equilibrium temperature when dry-bulb temperature is at its limiting value

by air drawn through the cooling chamber to a temperature which will deviate but little from the adiabatic saturation temperature before it enters the saturating chamber, tube A, through a connecting tube at C. This residual difference between the temperature of the make-up water supplied to the adiabatic saturator and the adiabatic saturation temperature will cause negligible errors in the equilibrium temperature reached at the outlet of the saturating chamber.

Should check readings only be required, it would be sufficient to wet the sponge beforehand. The sponge in a first prototype instrument, manufactured for testing purposes, contained about 50 cc of water. As the amount of water which must be evaporated under normal conditions is small (see Fig. 4), it follows that the instrument can operate for several hours without drying out. In this case the temperature of the surplus water in the sponge will first be cooled to the adiabatic saturation temperature and from then onward it can be assumed that the make-up water is supplied at a temperature equal to the adiabatic saturation temperature. Tests have shown that with the experimental instrument tested, equilibrium conditions are reached within a period of approximately five min.

When evaluating humidity from dry-bulb and adiabatic saturation temperatures, the commonly-used psychrometric equation (Equation 1) may be used. Values of the "psychrometric constant," A, calculated for different dry-bulb and adiabatic saturation temperatures, are shown in Fig. 6. It can be seen that A is not a constant but primarily a function of the adiabatic saturation temperature. The graph, Fig. 6, is applicable to conditions where the difference between the dry-bulb and adiabatic saturation temperature varies from zero to 20 F, i.e. for saturated airstreams as well as air having a low humidity. As the scatter of the plotted points is exceedingly small, no detectable errors will arise when using a "psychrometric constant" corresponding to a certain adiabatic saturation temperature, irrespective of the dry-bulb temperature.

For instance, the value of the psychrometric constant, for saturated air at 80 F, is 3.546×10^{-4} (F^{-1}), from Table II. For a dry-bulb temperature of 100 F and adiabatic saturation temperature of 80 F, the psychrometric constant becomes 3.542×10^{-4} (F^{-1}). For these conditions the difference in the values for the psychrometric constant will cause a difference of less than 0.03% in the vapor pressure when calculated from Equation (1) for an atmospheric pressure of 25.78 in. Hg.

Experimental investigation – For the experimental investigations conducted to prove the practicability of the instrument and to test its performance, air of known dry-bulb and dew point temperatures was obtained by means of an apparatus described in detail by Grant.[9] The diagrammatic layout of this apparatus is shown in Fig. 7.

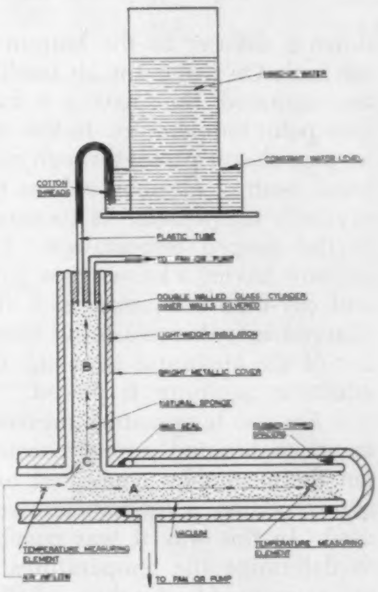


Fig. 5 Continuous reading adiabatic saturator

It consists of a tank containing about 20 gal of water, the temperature of which may be controlled. The water is circulated through a venturi where air is entrained with the water and carried

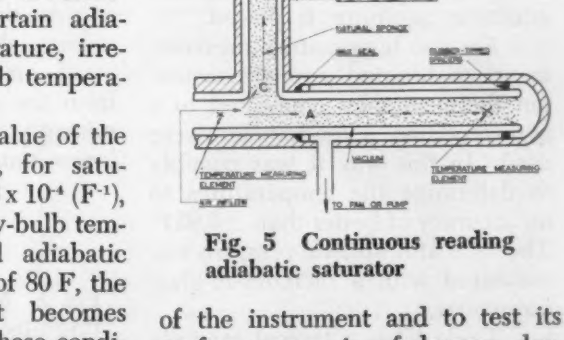


Fig. 6 Values of the psychrometric constant calculated for different dry-bulb and adiabatic saturation temperatures

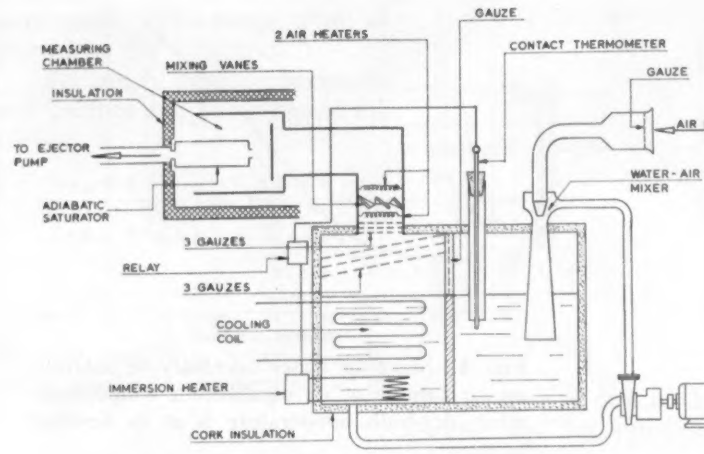


Fig. 7 Layout of the humidity control apparatus

down a diffuser to the bottom of the tank. On rising, the air bubbles are saturated, thus having a fixed dew point temperature. In the outlet pipe the air passes through electrical heating elements where the dry-bulb temperature is increased to the desired temperature. The air, now having a known dew point and dry-bulb temperature, is discharged into the measuring chamber of the apparatus in which the adiabatic saturator is placed.

For the temperature measurements, calibrated copper-constantan thermocouples connected to a high-precision potentiometer were used. In this way it was possible to determine the temperatures to an accuracy of better than ±0.05 F. The total atmospheric pressure was measured with a mercury-in-glass barometer.

Table VI is a typical example of a test series. The duration of the test was about 6 hr and the instrument functioned continuously. The response time of this instrument after a change in the state of the incoming air sample is about 5 min for a ventilation rate of approximately 120 fpm and is illustrated in Fig. 8.

In calculating the dew point from the test results, Equation (1) was used, viz.

$$e = e'_s - AP(\theta - \theta')$$

where P is the prevailing atmospheric pressure.

The psychrometric constant, A, however, was calculated using a value of 25.78 in. Hg, which is a good mean value for Pretoria. The difference between the prevailing atmospheric pressure and the mean value of 25.78 in. Hg was too small to cause a detectable error in the calculated value of the dew point temperature.

Dew point temperatures calculated from the measured dry-bulb and equilibrium temperatures agreed within ±0.1 F with the known dew point temperatures in the test chamber of the special humidity apparatus irrespective of the controlled humidity of the air sample.

As the insulation around the mixing chamber is not ideal, a small amount of heat will flow from the surrounding air into the mixing chamber which is at a lower temperature level. Thus, a fraction of the heat necessary to evaporate the water will be obtained from an outside source. To evaluate the influence of this heat gain, the velocity of the air flow through the mixing chamber was increased from approximately 25 to about 250 fpm.

The wet-bulb temperatures, measured with a standard Assmann psychrometer (while the test conditions remained constant) were compared with the equilibrium temperatures reached in the mixing chamber of the adiabatic saturator at increasing rates of ventilation. The difference between the respective temperatures is plotted as a function of the ventilation rate through the adiabatic saturator in Fig. 9. Once the ventilation rate through the adiabatic saturator exceeds 50 fpm, the difference between the measured temperatures remains constant. The wet-bulb temperatures, as measured with a standard Assmann psychrometer, tend to be too high, as the rate of ventilation over the wet-bulb thermometer is, in most cases, much lower than the assumed value of 1000 fpm. Thus, increasing the rate of ventilation through the adiabatic saturator beyond 50 fpm only increases the speed of response of the instrument

Fig. 8 Speed of response of the adiabatic saturator

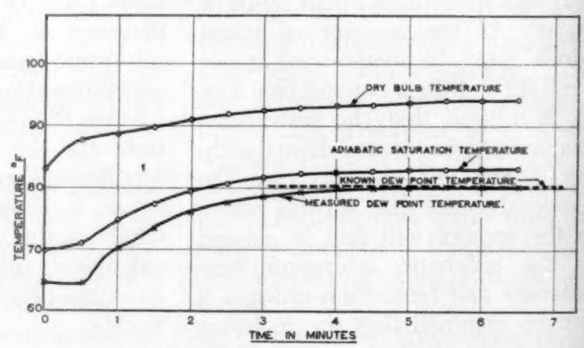


Table V. Errors in the Measured Equilibrium Temperatures and Calculated Dew Point Temperatures When Make-up Water Supplied is Five Times the Amount Evaporated

1 Dry-bulb Temp	2 Adiab. Sat. Temp	3 Equil. Temp	4 (3-2) Error	5 Measured Dew Point	6 True Dew Point	7 (5-6) Error
85	80	80.02	+0.02	78.68	78.66	+0.02
90	80	80.09	+0.09	77.39	77.26	+0.13
100	80	80.42	+0.42	74.93	74.31	+0.62
120	80	81.71	+1.71	70.52	67.48	+3.04
70	60	60.14	+0.14	55.04	54.79	+0.25
80	60	60.63	+0.63	49.87	48.53	+1.34
100	60	62.54	+2.54	38.83	29.80	+9.03
50	40	40.24	+0.24	29.52	28.86	+0.60
60	40	40.90	+0.90	14.38	9.58	+4.80

without influencing the equilibrium temperature reached within the mixing chamber.

The time taken to indicate 95% of a change from 80 F to 60 F wet-bulb temperature is shown as a function of the ventilation rate in Fig. 10.

As the resistance to air flow through the sponge is high, see Fig. 11, the pressure drop across the adiabatic saturator at high ventilation rates has to be deducted from the prevailing atmospheric pressure when calculating dew point temperatures using the psychrometric equation.

CONCLUSIONS

1. The adiabatic saturator developed is capable of measuring the adiabatic saturation temperature to a high degree of accuracy. Once this is known, the absolute humidity can readily be found by calculation or from tables. It is an absolute instrument.
2. This instrument can easily be adapted for continuous recording and controlling of humidity.

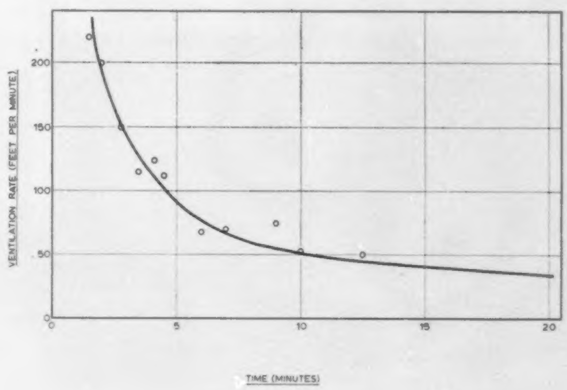
ACKNOWLEDGMENTS

The adiabatic saturator was developed in the laboratories of the National Mechanical Engineering Research Institute. The author extends his appreciation to S. J. P. Joubert for his criticism and advice.

Table VI. Measured Dew Point Temperatures as Compared with Controlled Dew Points

1 Dry-bulb Temp F	2 Equilibrium Temp F	3 Measured Dew Point F	4 Controlled Dew Point F	(3-4)
119.2	88.0	80.6	80.7	—0.1
119.0	88.2	81.0	81.1	—0.1
117.7	87.8	80.7	80.7	0
114.4	86.7	79.9	79.9	0
114.1	86.6	79.8	79.9	+0.1
108.4	86.5	81.2	81.1	+0.1
108.4	86.5	81.2	81.2	0
105.0	85.5	80.8	80.7	+0.1
99.2	84.4	80.7	80.8	—0.1
88.0	83.2	82.0	82.0	0
87.6	82.7	81.5	81.5	0
87.6	82.8	81.6	81.3	+0.1
84.9	82.0	81.2	81.3	—0.1
84.8	82.0	81.3	81.2	+0.1

Fig. 10 Time taken to indicate 95% of a change in wet-bulb temperature of 20 F (from 80-60F) as a function of the ventilation rate through the adiabatic saturator



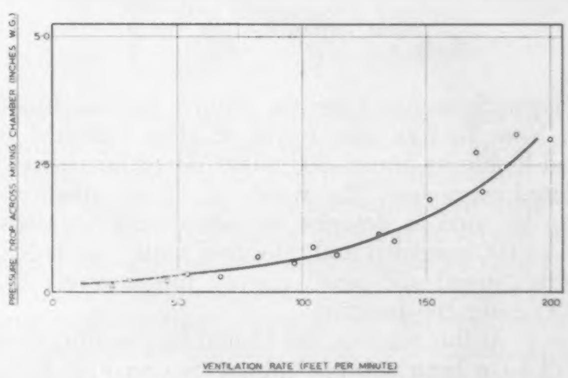


Fig. 11 Ventilation rate through the adiabatic saturator as a function of the pressure drop across the instrument

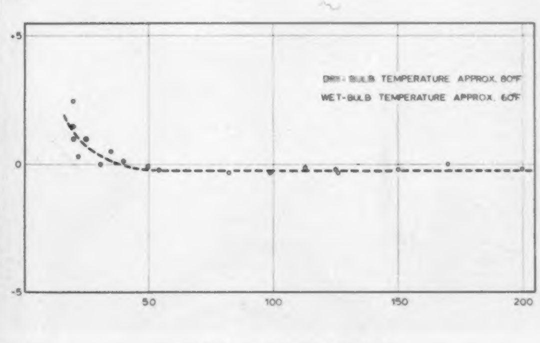


Fig. 9 Difference between the wet-bulb temperature as measured with a standard Assmann psychrometer and the equilibrium temperature as indicated by the adiabatic saturator as a function of the ventilation rate through the adiabatic saturator

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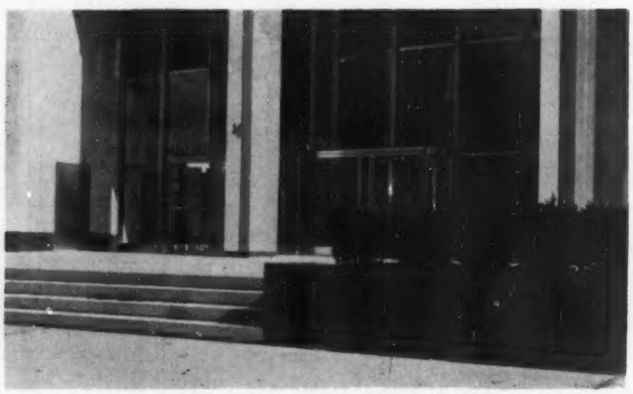
With ASHRAE

at the new United Engineering

J. H. CANSDALE
Assistant Secretary



Main entrance—47th Street

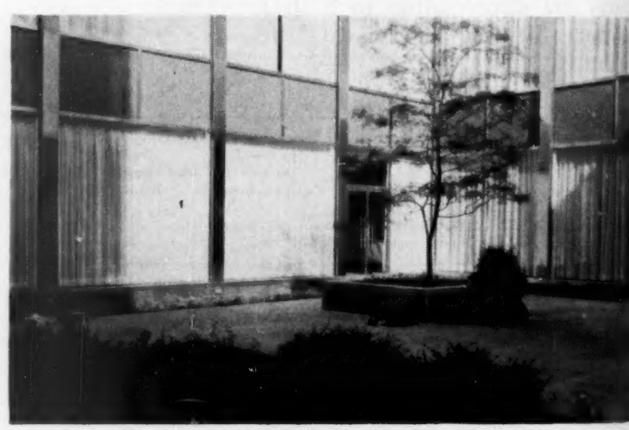
Diagonally across from the United Nations Building in New York, a slim tower of glass, stainless steel and limestone houses 600 rather dazed but somewhat elated employees. The word . . . if any single word can be used to describe the mood and reactions of these 600 uprooted and relocated souls . . . is NEW! Sight, sound and smell convey this newness to all who enter the building.

At this reading, the United Engineering Center will have been occupied about two months. For the first several weeks, headquarters employees of 21 engineering societies, with a combined membership of more than 300,000, shared the elevators with workmen and dodged the scaffolding and appurtenances involved with last-minute painting and decorating.

The paint is NEW; the plantings are NEW and still untouched by city smoke and grime; the elevator system is NEW and the first few rides presented a challenge to the daring; furnishings are NEW; the cafeteria line still fascinates and commuting to work has introduced NEW travel routines and methods of transportation to many. Most perceptible, the feeling of UNITY is NEW.

Reference library will provide ready access to 180,000 volumes; 3400 periodicals

Outside the UEC cafeteria is this quiet close



For the first time in the history of American engineering, more than a quarter of a million people are conscious as never before of a single, united purpose—to contribute to the advance of civilization through the skills and achievements of engineering. The United Engineering Center is the Free World's greatest assembly of Societies in many of the profession's branches.

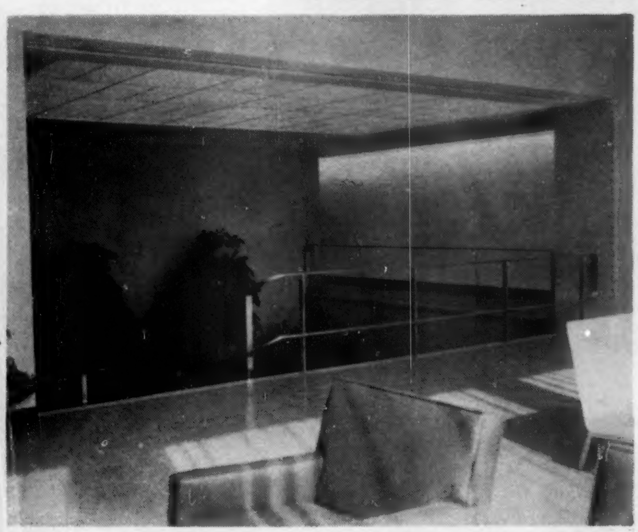
If there is no revolutionary concept in the edifice which houses these Societies, it is because engineering, right along, has provided new buildings with the latest and best, so that when time came to create its own headquarters, the improvements developed

Center

ASHRAE's new home is the most recent addition to New York's fabulous skyline

United Engineering Center

UEC reception area



for others were incorporated in the Center. It is a testimonial to the profession's attention to the competence and quality of myriad buildings that its best service to itself was to do what it has been doing for others.

The entrance lobby of this 20-story building is marble-walled and terrazzo-floored. Corridors lead to meeting, dining and semi-public rooms and to four automatic passenger elevators. On the East side of the lobby, there is a 6234-sq ft exhibition hall.

The Center will be in constant use by members of all the tenant societies for meetings of various sizes and accommodations have been designed accordingly. The main auditorium, seating 434 persons, may be partitioned into three separate rooms. There are complete facilities for film and lecture presentations. Other meeting rooms are available, with luncheon and dinner accommodations when meetings are held at meal periods.

Acoustical ceiling and flexibly partitioned areas are features of the auditorium

What is probably the world's largest free public engineering library, the Engineering Societies Library, is on the second floor. It has an 81-seat capacity in the reading room.

An employees' cafeteria of 200 seats is located on the ground floor looking out on a sunken, landscaped court. Meeting-dining rooms include one for 135 persons and two smaller ones for 20 each.

ASHRAE occupies approximately 5000 sq ft on the fourth floor. Our neighbor on this floor is the American Institute of Industrial Engineers. The newness of unity is especially significant for the ASHRAE staff of 40. For the first time since the merger, all personnel are together in one quarter. Within the brief time that the staff has been together, there is clearly in evidence a marked improvement in efficiency and communications.

ASHRAE reception area

From the ASHRAE entrance. Office at the far right is that of the Executive Secretary. Open doorway is to the office of the Associate Secretary and of the Technical Secretary. Partitioned offices are for staff members.

One of the primary objects of the United Engineering Center is to offer greatly expanded service and convenience to member engineers throughout the United States and Canada, as well as in many other countries. This is also the objective of the ASHRAE staff. The building belongs to all members of all the societies and our facilities and hospitality are at your command. You are cordially invited to visit the Center whenever you are in New York. We are justifiably proud of this building, not just as a pleasant place in which to work, but as a symbol of the progress and unity of the engineering profession.

UEC reception area

St. Louis Plans Ahead for 1962 Semiannual Meeting

January 29—February 1

Planners Waites, Stout, Norris, Myers, Evans, Cuba and Jaeger at a recent meeting where procedures for a full technical and social program were discussed.

W. F. Halpin, Jr.

J. G. Hamm

George W. Myers
General Chairman

Bruce L. Evans
Vice Chairman

Frank G. Jaeger

Finance

Wayne L. Stout	Chairman
Clarence K. Meyers	Vice Chairman

Sessions

Frank J. Zellner	Chairman
R. A. Behrmann	Vice Chairman

Banquet

John G. Hamm	Chairman
E. E. Carlson	Vice Chairman

Entertainment

W. F. Halpin, Jr.	Chairman
John D. Rosebrough	Vice Chairman

Luncheon

Bruce L. Evans	Chairman

Transportation

Clyde A. Durphy	Chairman
C. F. Barmeier	Vice Chairman

Reception & Hospitality

R. R. Waites	Chairman
W. David Bevirt	Vice Chairman

Publicity

Joseph F. Cuba	Chairman
Donald E. Hanlon	Vice Chairman

Ladies

W. P. Norris	Chairman
Edward T. Clucas	Vice Chairman

Special Events

Frank G. Jaeger	Chairman

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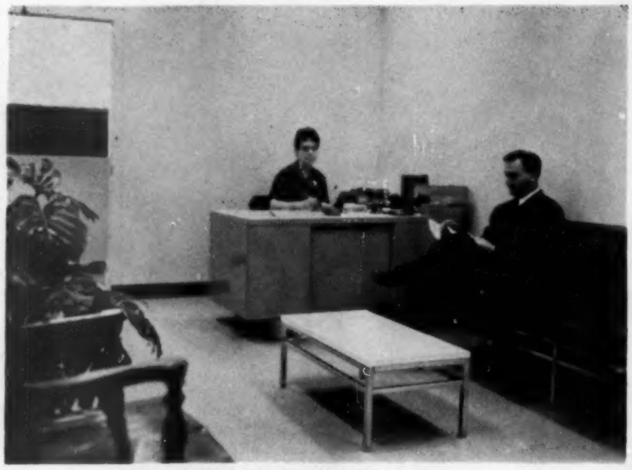

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Meetings ahead

November 5-7—National Frozen Food Association, National Convention and Exposition, Bal Harbour, Fla.

November 6-10 — National Warm Air Heating and Air Conditioning Association, 48th Annual Convention, Chicago, Ill.

November 6-11—Experimental Center for Refrigeration, Fourth General Assembly, Valencia, Spain.

November 12-15 — Air Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

November 16 — National Electrical Manufacturers Association, 35th Annual Meeting, New York, N.Y.

November 26-December 1—American Society of Mechanical Engineers, Winter Annual Meeting, New York, New York.

November 28—ASHRAE-ASME co-sponsored Psychrometrics Meeting, New York, N. Y.

November 28-30 — Building Research Institute, Fall Conferences, Washington, D. C.

January 29-February 1—American Society of Heating, Refrigerating and Air Conditioning Engineers, Semi-annual Meeting, St. Louis, Mo.

February 9-11—Air Conditioning and Refrigeration Wholesalers, Annual Convention, Los Angeles, Calif.

February 12-15—12th Exposition of the Air Conditioning, Heating and Refrigeration Industry, Los Angeles, Calif.

March 3-8—National Association of Frozen Food Packers, Chicago, Ill.

March 19-21 — National Electrical Manufacturers Association, Second National Electric Comfort Heating Symposium and Exposition, Chicago, Ill.

May 8-11 — Mechanical Contractors Association of America, 73rd Annual Meeting, Philadelphia, Pa.

May 22-23—American Institute of Electrical Engineers, Domestic Appliance Conference, Columbus, Mo.

June 4-7—Institute of Boiler and Radiator Manufacturers, Annual Meeting, Absecon, N. J.

News of ASHRAE members

New jobs

Calvin L. Kleinschmidt has joined the Trane Company Detroit sales office as Sales Engineer. A graduate of the University of Missouri, he brings to the company eight years' experience in the air conditioning field.

Marion K. Cox, previously Supervising Mechanical Engineer for Jules P. Channing Associates, becomes sales engineering representative for Acme Industries, Inc.

Herman B. Castillo has been appointed Chief Engineer of Temprite Products Corporation. With Temprite since 1956, he succeeds **Frank O. Graham**, who is retiring and will be retained on a consulting basis. Prior to 1956, Mr. Castillo had been an instructor of refrigeration engineering at Lawrence Institute of Technology and at the Detroit Air Conditioning Institute.

Chris N. Cuddeback, a 1957 mechanical engineering graduate of Cornell University, joins Trane Company as a sales engineer in the Bethesda, Md., office. Appointed sales engineer in the Wilmington, Del., office is **Arnold A. Granke.**

Jacob Koton and **William J. Donovan** have announced the formation of a new consulting engineering firm, Koton and Donovan, with offices in West Hartford, Conn. They will specialize in mechanical and electrical engineering in the air conditioning, refrigeration, heating, plumbing and electrical fields. Graduated from the Munich Institute of Technology, Jacob Koton was Manager of Production at the Middle East Tube Company and a lecturer at Technion Haifa. He has also been an instructor at the University of Hartford and headed Jacob Koton Associates, a mechanical and electrical engineering company. William J. Donovan was graduated from Webb Institute of Naval Architecture and received a Master of Science degree from Massachusetts Institute of Technology. Formerly Director of Engineering for Bush Manufacturing Company, he more recently headed Donovan Associates, a manufacturers' representative organization.

DONOVAN

KOTON

William Dedmond of Montreal Chapter is now Manager of Hart & Cooley. Another Chapter member, **John Goodenough,** has moved from Hart & Cooley to Powers Regulator Company.

David McDowell, who has left Powers Regulator Company to join Public Service Company of North Carolina, and **John Speer,** who is opening a Raleigh, N. C., office for American-Standard Industrial Div, are two South Piedmont Chapter members assuming new posts.

Edward J. Brown, former Research Assistant Professor of Mechanical Engineering at the University of Illinois, has been appointed Research Engineer at the Titus Research Laboratory, Titus Manufacturing Corporation. While with the University, from which he received his B.S. and M.S. degrees in 1954 and 1959, respectively, he was principal investigator on a number of projects, including an investigation of dampers used in large duct systems and the residential year round air conditioning project of the National Warm Air Heating and Air Conditioning Association. He has contributed a number of articles to technical publications.

Frank L. Vaughn and **Alan Owens** have been promoted to District Managers of Sporlan Valve Company. Both had been sales engineers prior to their new appointments.

John Liebermann, with Ranco, Inc., since 1949, has been appointed Chief Engineer. He is a graduate of the University of Vienna and the University of London. Named Product Manager, **Thomas I. Syfert** is a graduate of Ohio State University and joined Ranco in 1951.

Carlisle D. LaGasse has formed his own air conditioning contracting firm, Southwest Air Conditioning, Inc. Formerly a partner in an air conditioning contracting company, he was for four years a district sales manager for Westinghouse Electric Corporation. He is a graduate of the University of California.

Gunnar C. F. Asker has been appointed Vice President and Assistant to the President of Cargocaire Engineering Corporation. One of the co-founders of Desomatic Products, Inc., now a Div of Atlantic Research Corporation, he was President of that company until February 1961. A graduate of the Royal Institute of Technology in Stockholm, Sweden, he is a past-President of the former Washington Chapter of ASHRAE.

Charles D. Wright, named Southern Regional Manager, and **Robert F. Logsdon**, appointed Manager of Engineering for the Reed Div, are two new promotions announced by American Air Filter Company. Re-assigned were **Don J. Gonzalez**, named Coordinator of Engineering, and **Karl I. Westlin**, appointed Manager of Product Engineering.

Retirements

Walter O. Walker, Fellow ASHRAE, is retiring as Dean of Research and Industry and Professor of Chemistry at the University of Miami. He will continue his work in refrigeration research as Director of the Industrial Chemical Research Laboratory.

Necrology

Leonard N. Reed, Vice President of Davis Refrigeration Company, died on September 12. Associated with the company for 20 years, he had served previously as an engineer.

Theodore I. Messer, recently elected an Affiliate of ASHRAE, died on August 24, 1961, at the age of 57.

Stanley C. Smith, deceased, had been Senior Mechanical Engineer with the Department of National Defense, A.F.H.Q. in Ottawa, Ontario, Canada.

Frank C. Winterer, a Life Member (elected 1920), died recently. He was 67.

Robert J. Waalkes died on September 10 at the age of 40. A Product Engineer with Hart & Cooley Manufacturing Company, he had served in Western Michigan Chapter as a member of the Board of Governors (1952-53, 1959-60), Treasurer (1955-56), Secretary (1956-57), Vice President (1957-58) and President (1958-59).

Harold A. Brangs, who died on August 11, had been associated with L. J. Wing Manufacturing Company.

Byron L. Casey was District Sales Manager for Ilg Electric Ventilating Company at the time of his recent death. He was a life Member.

Shelby B. Evans, Chief Engineer with the T. Eaton Company, died recently at the age of 66.

Charles F. Eveleth, deceased Life Member, had held office in Philadelphia and Cleveland Chapters and served on many technical and code committees of the Society, as well as on the Committee on Research.

G. H. Lehle, Consulting Engineer, died on August 29. He was a Life Member.

Herbert S. Moore, affiliated with Ontario Chapter since 1923 (Vice President and President 1930-31), died recently. A Life Member of ASHRAE, he had served also on the Nominating Committee.

Elmer G. Smith, Professor of Physics at Texas A. & M., died recently at the age of 64.

Edwin C. Evans, a Life Member, died recently at the age of 84. Joining the Society in 1918, he was instrumental at that time in increasing the membership of St. Louis Chapter, and served later as President of Philadelphia Chapter.

David J. Wood, National Sales Manager for Frick Company, died recently.

Others are saying—

research and development . . . is the fastest growing sector of U. S. industry, with an estimated $3.5 billion spent for this purpose in 1961. More than 5400 non-government research laboratories handle fundamental and applied research. Considered in this article are budgeting for research, organization and staffing of a laboratory, research aims and value of independent inventors. *News Front, September 1961, page 10.*

resistance by scientists . . . to scientific discovery has been neglected by scholars investigating the resistance to scientific discovery of other social groups. However, it is a constant phenomenon with specifiable cultural and social sources. Investigation of resistance to scientific discovery can lead to a greater knowledge of the sources of acceptance, so that the former may be reduced and the latter increased. *Scientific Manpower 1960, May 1961, page 36.*

thermoelectricity . . . cited as the first of static energy converter techniques to move from the laboratory into commercial application, is still far from being competitive for heavier duty applications. However, the importance of thermoelectricity as a static power generator is assured by the versatility resulting from its satisfactory utilization of any available heat source. Other advantages are quiet operation, no maintenance, long life, simplicity and potential applications for small amounts of cold. *Industrial Research, October 1961, page 28.*

control of environment . . . including temperature, humidity and pressure, in contamination control areas depends primarily on proper design and selection of mechanical equipment, such as control and air distribution devices. Characteristics of various systems are discussed and diagrammed herein. *Air Engineering, August 1961, page 20.*

BURT LOMAX, JR. HEADS REGION VII
Though listed correctly previously (September, page 84) Regional Director Burt Lomax, Jr. was not so indicated in the October issue (page 87 listing of Region and Chapter Officers). Mr. Lomax's present term is June 1961 to June 1962.

ASHRAE OFFICERS, DIRECTORS, COMMITTEEMEN

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J. Everetts, Jr.
First Vice President
John H. Fox
Second Vice President
Frank H. Faust
Treasurer
John E. Dube
Executive Secretary
R. C. Cross

Executive Secretary Emeritus
M. C. Turpin

Past Presidents
R. H. Tull
D. D. Wile

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George Linskie
J. G. Woodroof (IV)
James Downs (V)
L. K. Warrick (VI)
Fred Janssen (IX)

June 1960—June 1962
W. S. Harris
W. L. McGrath
J. W. May
Axel Marin
G. B. Rottman
V. D. Wissmiller
W. J. Collins, Jr. (VIII)
T. J. White (X)

June 1961—June 1962
Burt Lomax, Jr. (VII)

June 1961—June 1963
P. R. Achenbach

June 1961—June 1964
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W. B. Morrison
P. W. Wyckoff
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C. H. Flink
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E. R. Searles

EXECUTIVE

J. Everetts, Jr., *Chairman*
John H. Fox
Frank H. Faust
John E. Dube
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D. A. Solley
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QUALITY

(*Continued from page 42*)

sumer can recognize it. To sell on a quality basis you will have to find more effective ways of convincing consumers that your product has it.

Underwriters Laboratories by its UL approval has meant "electrically safe" to many consumers. Would it be too Utopian to conceive of the consumer's accepting a properly defined and suitably policed label for overall quality on an appliance? There are Government quality grades for meat and other food products. The AGA has the beginnings of such a system. Many manufacturers buy components on this basis. Can it be made to work for the Appliance Industry's consumer products? And to keep initiative from being stifled, why not several degrees of quality, so that an enterprising manufacturer can make a Grade 1, a Grade 2 and a Grade 3 quality appliance, each at a different level of quality for a different segment of the market?

If the grade label were then promoted for what it was, explained to the consumer in a noncompetitive way by the whole industry, and if it were an honest effort, I believe it would provide the necessary persuasion. Competition could then be around style, special features and price. And now the consumer could truly see what he was buying and what he was paying for it.

A far less controversial, though not an easier, solution is available for the production and the servicing problem. Many appliance manufacturers, moved by the recent drop-off in sales and by the bad image of the industry being created increasingly by experiences such as I and others have described, have already begun to do something about them. In the area of production, the solution lies in better quality control in the broadest sense: assuming a good design concept, all components must be inspected rigorously before use, workmanship on the assembly lines must be controlled at high levels, and, most important, the final product must be checked thoroughly for overall safety and performance.

Shipping practices must be watched so that the product arrives at its final destination in the condition originally intended. Increasing the effectiveness of the feedback system from service organizations, and using the information so obtained, can provide another invaluable method for better control of production problems.

The problem of service has also been attacked by some manufacturers. Some have approached it by improving the durability and trouble-free operation of the appliance in the design and production stage. Others have tried to design for ease of repair. Improved parts inventory and service, tightening the service organization for faster, more courteous, considerate and competent handling of customer complaints have also been used. And some manufacturers have assumed greater responsibility for service than heretofore. But if these are to be effective, the consumer must be convinced that they will provide him with minimum breakdown or low-cost, competent service. But how can this be done?

Nothing will convince a consumer that a manufacturer means what he says better than a foolproof guarantee. (And I don't mean the kind of guarantee which is designed to reduce the manufacturer's liability.) If, indeed, the manufacturer has solved the service problem satisfactorily, the proof he can offer a consumer is a good guarantee.

All of this, of course, is easier said than done. It is easier to point out directions than to plow the road to the end. I cannot, in good conscience, close this overview of these difficult problems without calling attention to a current practice which presents a large roadblock, making any serious effort to solve these problems many times more difficult than it needs to be. I mean the practice of the Annual Model Change.

Putting aside all the other reasons which may be cited for its discontinuance, it would make design, production and servicing infinitely easier. With enough time, models can get careful design consideration and then be durability and field tested before they are put on the market. I need not remind you, I am sure, of the consequences of the annual model change in these areas. With more time, also, there is time to train production and servicemen more effectively (not to mention salesmen who often know far too little of the product's true virtues). The economics of mass production and the extra time make it more feasible to manufacture, supply and stock parts earlier, more widely and for longer periods. All in all, at least in terms of the problems we have been discussing, the demise of the Annual Model should be welcomed as perhaps the simplest and largest single step that can be taken.

What ASHRAE Regions and Chapters are doing

Headlining their September meetings, which began the 1961-62 season, Chapters scheduled such diverse programs as discussions of thermo-electric refrigeration, air distribution, compressors and studies of wet and dry bulb temperatures.

TUCSON . . . Planned for the September 12 meeting was a field trip to the University of Arizona, which was to consist of a conducted tour through the central heating and cooling plant.
who's doing what . . . Committee Chairmen for the new season are: Arnold G. Greenheck, Membership; James S. Blackmore, Program; Casey Jones, Attendance and Reception; T. Thomson, Technical; Richard E. Joachim and H. B. Glover, Special Events; and John O'Hare, Publicity.

SOUTHERN CALIFORNIA . . . "Evaluated Weather Data for Cooling Equipment Design" was the topic of September speaker Loren W. Crow, Consulting Meteorologist. Illustrated, his talk dealt with his studies of dry and wet bulb temperatures as related to time.

At the Board of Governors meeting on July 31, Presidential Member A. J. Hess reported on the United Engineering Center.

CENTRAL OHIO . . . Changes in the Society were discussed at the September 18 meeting by Presidential Member R. H. Tull, giving a breakdown of Society expenses. Questions were answered and methods of financing local Chapters were discussed.

"Thermodynamic Problems in Missile Environment" was the subject of guest speaker Charles Jones of Ohio State. Figures on problems of missile propulsion and re-entry were presented.
who's doing what . . . Emil Stluka and G. H. Reverman will represent this Chapter at the Regional Meeting.

DALLAS . . . Glen Scott, Public Information Supervisor for Southwestern Bell Telephone Company, was guest speaker at the August 20 meeting. Announced for September was a program on Civil Defense.
who's doing what . . . George Meffert reported on technical aspects of the Denver meeting.

NORTHEASTERN OKLAHOMA . . . First meeting of the season, held September 18, was on sound control and air conditioning environment. Guest speaker J. W. Spradling of Carnes Corporation covered the noise level of air moving equipment and its relationship to the type and size of the structure being conditioned. Sound control measuring devices were discussed, together with problems of evaluating a unit using information provided by such instruments.
who's doing what . . . J. W. Langston, a student at the University of Tulsa, is this year's recipient of the Chapter's H. W. Meinholtz Scholarship, made annually to an upper-class student in the School of Engineering.

CENTRAL INDIANA . . . Topic of a panel discussion held at the September 12 meeting was "Selection, Installation and Servicing of Compressors for Refrigeration and Air Conditioning". Speaking were: W. F. Frieje, Moderator; E. Hunter of Duncan Supply Company; W. Kennard of Industrial Weathermaker; and C. Hottle of F. H. Langsenkamp Company. Written questions submitted by the members were answered and discussed.

PUGET SOUND . . . Speaking at the September 12 meeting were R. Kirkwood and R. Lux of John Graham and Company. Discussed was the "Century 21 Space Needle," with mechanical considerations covered in detail.
who's doing what . . . Named as Chairman of committees for the Region X Meeting were: C. A. Pangborn, Program; B. Pride, Finance; J. Salsbury, Registration; T. Allison, Hotel; K. J. Barnet, Entertainment; K. Gablin, Banquet; C. Robson, Transportation; C. McMillan, Publicity; R. Lux, Printing; and R. Stern, Reception.

HOUSTON . . . Scheduled for the September 26 meeting was a program of addresses by past-Presidents of the Chapter. To speak were: R. F. Taylor, "The Founding of the Houston Chapter"; A. Barnes, "Heating, Past and Future"; A. Chase, "Cooling and Air Conditioning of the Past and Future"; and H. Broadwell, "Low Temperature Refrigeration of the Past and Future."

INLAND EMPIRE . . . Prior to the main talk of the evening at the September 11 meeting, T. Brand, J. R. Morris, R. B. Campbell and F. Jenkinson spoke on various subjects.

Accompanying his talk by slides, George Stabenow, Vice President in Charge of Engineering for International Boiler Works, spoke on "High Temperature Hot Water Systems." Schematic diagrams pointed out aspects of each specific design.

FORT WORTH . . . President J. R. Blanke announced at the September 20 meeting that the October meeting would be a joint meeting with Dallas Chapter.

Speaker of the evening was Terrell J. Small, covering "Converting Water-Cooled Equipment to Air-Cooled." Assisting him was Carl Yarborough, who showed slides to illustrate certain phases of the

discussion. Factors entering into the process of calculating the size of an air-cooled condenser needed to replace a cooling tower and water condenser were detailed. Cited by speaker Small as necessitating conversion of water-cooled systems were such problems as corrosion, water shortages and maintenance.

who's doing what . . . G. F. Smith, J. R. Blanke, D. Reid, L. Tye, J. Chatmas, S. Lake, T. Romine and C. Zahn are members of the newly-appointed committee on the UEC Fund Raising Campaign.

OREGON . . . Two Chapter members, Ray Chewning and Omer Jacobson, were scheduled to discuss central heat pumps at the season's first meeting, September 14. Background, criteria for feasibility, limitations of concepts and water-to-water applications were to have been the subject of speaker Chewning, followed by speaker Jacobson covering details of a specific air-to-water installation at Mabel Rush Elementary School.

ALAMO . . . At a meeting of the Board of Governors, held September 15, the following Committee Chairmen were named: R. E. Reese, Membership; J. H. Powell, Program; K. A. Monier, Reception; O. W. Schuchart, Technical; G. S. Smith, Attendance; and A. J. Rummel, Publicity. R. E. Reese and L. H. Horner were named Region Delegate and Alternate, respectively.

EL PASO . . . Presented at the Sept. 18 meeting was a talk on "Piping and Calculations—Heating and Cooling." Speaking was M. D. Goodwin.

EVANSVILLE . . . William G. Grief, Executive Director of Evansville's Future, Inc., spoke at the September 12 meeting on "Potentials for Future of the Evansville Area."

who's doing what . . . Vacancy in the Board of Governors caused by the resignation of I. Loeffler has been filled by appointment of J. Wellborn.

SOUTHERN ALBERTA . . . Reporting on the regional meeting, Chapter President N. J. Howes advised members on the need for technical committee representation.

September speakers were Kieth Goldsmith of Page Hersy Tubes and A. D. Henry of Crane Supply. Discussing various types of pipes and valves, they covered buttweld, continuous weld, electric resistance weld, seamless, lapweld and fusion weld.

CENTRAL ARIZONA . . . Guest speaker at the September 11 meeting was Robert S. Ash, Assistant to the President of International Metal Products Company, a Div of Edison-McGraw Company. He covered design of systems and special applications for evaporative cooling.

NEW MEXICO . . . Appointed at the September 12 meeting were the following Board Representatives from committees: W. Beale, Membership; P. Hood, Attendance and Reception; D. Paxton, Program and House and Special Events; L. Classen, Publicity; G. Sebree, Technical; and V. Stephens, Code. Committee Chairmen are: L. Meyers, Membership; J. Dion, Attendance and Reception; D. Braid, House and Special Events; L. Doremire, Publicity; R. Haines, Technical; and F. Bridgers, Code. Chapter Delegate and Alternate are D. Paxton and J. Desilets, respectively.

LOUISVILLE . . . First meeting of the 1961-62 season, held September 18, featured a discussion by Kenneth A. Mertz, Director of Engineering of the Air Moving Div of Torrington Manufacturing Company, on the problem of finding the proper air impeller for specific installations. This may be achieved by calculations involving pressure and flow coefficients and specific speed. Together with low power consumption, proper fans yield a low noise level.

Each type of impeller, he stated, has an operating range within which it gives optimum performance. Fan performance usually is correlated with a dimensionless parameter designated specific speed. Flow coefficient is defined as the ratio of the velocity of flow through the fan to the velocity at the blade tips. The pressure coefficient relates pressure output of the fan at peak efficiency to the velocity head available at the blade tips. Specific speed can be ascertained by calculation from the airflow requirements: air flow, pressure head required and desired impeller speed. From the values of the flow and pressure coefficients, basic air impeller proportions may be calculated.

NEW YORK . . . Preceding the September 28 meeting was a seminar on valves, featuring Thomas Maher of Walworth Company and George Gerza of Crane Company.

Relationship and coordination between the designer, engineer, contractor and client then was discussed by Maurice Mogulescu, President, Designs for Business, Inc.

who's doing what . . . A. Greenberg, Co-chairman of the Education Committee, announced a series of lectures on "Noise and Vibration" to be held shortly.

KANSAS CITY . . . Called upon to report at the September 5 meeting were: J. R. DeRigne, Chairman of the Membership Committee; D. M. Dart, Chairman of the Attendance Committee; and C. Shonfeld, Chairman of the Special Events Committee.

Covered in the talk of Professors Ralph G. Nevins and Paul Miller of Kansas State University, "Air Motion and Human Comfort", was design of

CHAPTERS REGIONAL COMMITTEE

COMING MEETINGS

REGION III, Hampton Roads Chapter (Norfolk), Nov. 4

REGION VII, Mississippi Chapter (Jackson), Nov. 6

proper air distribution for heating or cooling to provide adequate ventilation. Slides and charts illustrated the talk.

EAST TEXAS . . . Steering Committee for this new Chapter met on August 23 to discuss organizational details. Need for various committees was discussed by Chairman Walter Yeary, who called upon ASHRAE 2nd Vice President Frank H. Faust to detail the various functions of the committees. Committee appointments were made.

On September 28 the organization meeting was held, with 18 people in attendance. Walter Yeary requested that the chairmen of committees appointed at the prior meeting make their reports.

Ralph Ball of the Membership Commitee announced that the petition for the Chapter has a total of 27 names. W. A. Spofford, Program Committee, stated that ASHRAE President John Everetts, Jr., would speak at the October 24 meeting. Frank H. Faust read the model By-laws and discussion followed. Adoption of the By-laws was voted by the members. Chairman Loren Fletcher of the Nominating Committee presented the following as candidates for Chapter officers: Warren Spofford, President; Walter Yeary, Vice President; Robert Layton, Treasurer; William Arzberger, Secretary; and Charles Dubberley and Loren Fletcher, Board of Governors. The Chapter voted unanimously for adoption of this slate.

DAYTON . . . Principles of thermoelectric refrigeration were outlined briefly at the September 12 meeting by Dr. R. A. Bernoff, Director of Research, Materials Electronics Products Corporation. He showed how thermoelectric couples are made and utilized working models to demonstrate their use, both as heat pumps and as power sources. At present, he stated, relatively high cost and low efficiency of thermoelectric refrigeration materials limit their use to special applications where their small size, silent operation, durability and ease of reversing are major factors.

MEMPHIS . . . Program Chairman T. Turner announced at the September 18 meeting a projected series of research programs on system analysis, preliminary plans for which have been drawn up by that committee. Publication of papers from this series is intended.

who's doing what . . . New committee chairmen are: P. Hall, Membership; T. Turner, Program; H. Erb, Attendance & Reception; W. Wellford, Publicity; J. Wilson, Finance; General Danielson, Legislative; and J. Montgomery, Special Events. Chapter Delegate and Alternate are T. Bearden and P. Hall.

ST. LOUIS . . . Utilization of the oxygen supply for both breathing and cooling purposes in the Mercury space capsule was a feature of the capsule's system pointed out by September speaker Sanford McDonnell, Vice President of McDonnell Aircraft. Other details covered were construction and control system of the capsule.

who's doing what . . . Committee Chairmen for the 1961 ASHRAE Semiannual Meeting are: Joseph Cuba, Clyde Durphy, Bruce Evans, William Halpin, John Hamm, Frank Jaeger, George Myers, William Norris, Wayne Stout, Robert Waites and Frank Zellner.

SYRACUSE HOSTS REGION I MEETING

More than 150 engineers attended the Region I meeting on October 1-3 in Syracuse, N. Y. General Chairman was Stanley F. Gilman, Chairman of the ASHRAE Research and Technical Committee. An informal meeting of 24 delegates and alternates from the region was held October 1. Business meetings followed on October 2 and 3, presided over by Regional Director Perley K. Barker.

Other events included a luncheon at which ASHRAE President John Everetts, Jr., delivered an address on "The Road Ahead," and tours of Carrier Corporation and the new F. Ware Clary Junior High School, a fully air conditioned school.

Highlighting the Monday evening banquet were a short business meeting and a talk by Arthur J. Zito of General Electric Company. Toastmaster was William L. McGrath, a member of the ASHRAE Board of Directors.

Among others present for the meeting were National First Vice President John H. Fox and Past Presidents Walter A. Grant, Carlyle M. Ashley, Alfred E. Stacey and Logan L. Lewis.

Discussing plans for the Region I Meeting are Committee Chairmen Walter J. West, Walter H. Simmons, Stanley F. Gilman and John R. Schreiner

Here shown during a business session, ASHRAE President John Everetts, Jr., Regional Director Perley K. Barker and ASHRAE First Vice President John H. Fox were among 150 engineers in attendance.

MEMBERSHIP—

today's challenge, tomorrow's future

W. H. MULLIN
Chairman
Membership Development
Committee

Today's challenge is to introduce into our Society the new graduate, the young cadet engineers and technicians in training and other young people with the capabilities needed within our profession. If our Society is to have a vigorous and vital future, we must enlist students coming from our schools and assist them in developing their talents. To these young engineers belongs the future. If ASHRAE doesn't recruit them, another society will.

Young engineers are the potential leaders in our profession and in our Society. Enlist them now, help them grow and our record of achievement will live and be enhanced. Fail to meet this challenge and we can expect, at best, only mediocrity in our future Society activity.

No one person or single group can swell our ranks with the qualified people we need. Required are the combined efforts of every member—thinking, talking and acting. Will ASHRAE have a great future? The answer is yes . . . if you become an ASHRAE BOOSTER by recruiting at least one new member this year.

Don't sell ASHRAE short when informing a prospective member about the benefits of our Society. Certainly, every prospect will be interested in receiving our publications . . . the JOURNAL, GUIDE AND DATA BOOK, and other informative literature available, such as codes, standards, preprints, TRANSACTIONS, research reports and bulletins.

However, do not stop there! These are material benefits. The most important factor in our Society is "PEOPLE AND THEIR DEVELOPMENT". Tell your prospect what ASHRAE means to you, how it has helped you and many others to advance professionally.

Experience and knowledge of up-to-date technical developments are essential ingredients in professional advancement. ASHRAE, through its scientific and educational work, gives both the neophyte and seasoned engineer opportunities for gaining and broadening their experience through service on any of our many general and technical committees.

In addition to the Society publications and committee activities, local Chapter, Regional and National meetings provide unlimited scope for gaining more technical information.

Make sure that your prospect realizes that his ASHRAE membership will keep him well informed on developments in his specific field of interest as well as all aspects of heating, refrigerating, air conditioning and ventilating. It is the well-rounded engineer, today, who is needed to meet all challenges in the environmental industries.

ASHRAE membership furnishes a common ground for meeting and exchanging ideas with the top minds in our profession. The opportunity of associating and working with industry leaders is one of the great advantages of Society membership.

Everyone has a personal responsibility to advance the knowledge and technology of his profession to the fullest extent of his abilities. ASHRAE is the medium whereby each member's contributions may be promoted and put to the best possible use. Also, each man owes to himself and his "calling" the charge of maintaining and raising professional standards. What better way can this be done than through ASHRAE—your professional engineering Society?

Invite prospective members to attend Chapter and National meetings and, through Committee Chairmen, invite them to attend committee meetings. Active participation in the Society is a broadening and worthwhile experience. The inactive member does not get his money's worth. The old adage —"You get out of it what you put into it"—applies strongly to professional societies.

The officers of ASHRAE are anxious to have all members develop their talents to the fullest. Current members, who are not as active as they could be, are urged to increase their participation in Society affairs. MOST IMPORTANT, all members are asked to enlist new recruits to carry on our endeavors. Make your immediate goal the recruitment of another ASHRAE member.

Every Member an ASHRAE Booster

Candidates for ASHRAE Membership

Following is a list of 125 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before November 31, 1961 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

NIXON, G. M.†, Asst. Prof., College of Engineering, University of Bridgeport, Bridgeport.

Massachuetts

PERRY, G. H. JR., Design Engr., General Services Administration, Boston.
SHEPHERD, M. B. JR., Owner, M. B. Shepherd, Jr., Co., Pembroke.
STEBBINS, C. D., Sales Engr., The Trane Co., Boston.

New Jersey

WUNDERLICH, F. J.*, Private, U. S. Army, Fort Dix.
YODER, J. D., Industrial Hygienist, Humble Oil & Refining Co., Linden.

New York

ADDAMO, VINCENT, Designer, Albert Fentzlaff, Inc., New York.
BENZIGER, B. C., Sr. Htg. & Vent. Engr., N. Y. State Dept. of Public Works, Albany.
BERCH, CALVIN†, Engr., S. Gerber & Co., New York.
CONDE, D. F., Asst. Prof., Alfred State Technical Institute, Alfred.
CROWLEY, R. A.*, Branch Mgr., Barber-Colman Co., Albany.
HANLEY, E. O., Estimator Engr., Triangle Sheet Metal Works, Inc., New Hyde Park.
HUGGLER, J. B. JR., Mech. Designer, S. & M. Air Conditioning & Kiamesha Sheet Metal, Liberty.
LAPINSKI, JOHN, Pres., Levi Case Co. Inc., Schenectady.
MASSIMILLA, D. A., Sales Engr., Hi-Press Air Conditioning of America, New Pork.
MEURS, J. E., Pres. & Mgr., Meurs & Outtan, Inc., Albany.
ROGERS, R. W., Appl. Engr., Electromode Div., Commercial Controls Corp., Rochester.
ROMANZO, S. R., Estimator, Campito Plumbing & Heating, Inc., Albany.
SMELLIE, J. H., Supvsg. Engr., New York Telephone Co., Brooklyn.
VARNEY, C. L., Product Mgr., Johns-Manville Sales Corp., New York.
WESTCOTT, J. C., Engr. Assistant, Welch Grape Juice Co., Westfield.

REGION II

Canada

BOAG, R. L., Project Engr., James Howden & Co. of Canada, Ltd., Scarborough, Ont.
CRAIG, W. H., Consulting Engr., Toronto, Ont.
GIROUX, H. P. W., Gen. Mgr., Canadian Thermo Control Co. Ltd., Montreal, Que.
LANDYKE, R. A., Gen. Mgr., Industrial Oil Burner Service, Agincourt, Ont.
MARTIN, JOHN, Sales Engr., R. H. Dyson Co., Ltd., Winnipeg.
SCHELLENBERG, ERNEST, Mgr. A-C. Sales, Trane Co. of Canada, Ltd., Toronto, Ont.
ZACH, R. E.*, Chief Engr., Canadian Ice Machine Co. Ltd., Toronto, Ont.

REGION III

Maryland

DIONE, E. R., Purchasing Agent, Riggs Distler & Co., Inc., Baltimore.

Pennsylvania

AMAN, A. J., Sales Engr., Mercator Corporation, Reading.
BLACK, M. A. JR.*, Mgr. Frick Co., Waynesboro.
CULHANE, J. J., Appl. Engr., Philadelphia Electric Co., Philadelphia.
DELO, P. M., Mgr. Johns-Manville Sales Corp., Philadelphia.
GETTIG, D. U., A-C. Sales Engr., The People National Gas Co., Pittsburgh.
MCHALE, W. L., Sr. Research Engr., Robertshaw-Fulton Controls Co., King of Prussia.
SORENSEN, S. E., Cons. Engr., Robertshaw-Fulton Controls Co., King of Prussia.
WILLIAMS, C. E., Sales Engr., E. J. Deckman Co., Pittsburgh.

Virginia

ALLSBROOK, G. D., Design Engr., Vansant & Gusler, Norfolk.
WATSON, H. L.*, Sales Engr., Refrigeration Supply Co., Richmond.
WEISS, F. J.*, Prop., Frederic J. Weiss Co., Richmond.

REGION IV

Florida

RUSSELL, D. E., Cons. Engr., Jacksonville.

Georgia

HOLTERHOFF, RALPH JR.*, Sales Engr., Dunham-Bush, Inc., Decatur.

North Carolina

BROWNING, R. C., Cons. Engr., Raleigh.
CARPENTER, R. M.†, Sales Repr., The Trane Co., Charlotte.
TERRY, W. P., Vice-Pres. & Secy., Allan T. Shepherd Co., Charlotte.

REGION V

Indiana

BISHOP, A. J.*, Engr., Liniger Co., Inc., Marion.
BORROR, H. M., Mech. Engr., Eli Lilly & Co., Indianapolis.
BRIER, R. H., Estimator, Roland M. Cotton Co., Indianapolis.
GRINSTEAD, C. E.*, Staff Engr., Swartout Fabricators, Inc., Kokomo.
HOGLE, J. T., Estimating Engr., The Hogle Co., Indianapolis.
RAY, F. P., Contract Mgr., The Philip Carey Mfg. Co., Indianapolis.
ROTZ, R. A., Pres., J. M. Rotz Engineering, Indianapolis.
SWENSON, E. G., Sr. Engr., Mead Johnson & Co., Evansville.

Ohio

TAYLOR, J. L.*, Industrial Engr., Ohio Fuel Gas Co., Toledo.
WALL, R. J., Pres., The Wall-O'Connor Co. Inc., Toledo.

REGION VI

Illinois

JAEGER, U. R., Special Projects Mgr., Olin Mathieson Chemical Corp., East Alton.
KING, J. P., Sales Engr., Vilter Mfg. Co., Chicago.
MOREM, G. A., Engr., Rock Island Furnace Supply, Rock Island.
STOLNACK, E. M., Repr., McDonnell & Miller, Inc., Chicago.

Michigan

PRAKASH, RAJENDRA, Associate Professor, Michigan State University, Lansing.
WISNER, R. R.*, Design Engr., American Motors Corp., Detroit.

Minnesota

BRISCOE, D. D., Sales Engr., American Air Filter Co., Minneapolis.
MORRISON, J. F.*, Product Engr., Whirlpool Corp., St. Paul.

Wisconsin

DOYLE, F. W., Sales Mgr. Ind. Products, Perfex Corp., Milwaukee.

REGION VII

Alabama

ECHOLS, H. M., Appl. Engr., S. A. Brown & Co., Birmingham.
MORGAN, E. S. JR., Sales Engr., Barber-Colman Co., Birmingham.

Phillips, H. O. Jr., Appl. Engr., Barber-Colman Co., Birmingham.
Walston, D. W., Htg. & A-C Repr., Alabama Power Co., Mobile.

Kentucky

Cornett, W. L., Mech. Engr., Edward T. Hannan & Assoc., Paducah.

Missouri

O'Mara, W. T., Sales Repr., American-Standard, Kansas City.
Squibbs, G. G., Engr., Natkin & Co., Kansas City.

Tennessee

Hess, H. C., Design Draftsman, Burkhalter Hickerson & Assoc., Nashville.

REGION VIII

Arkansas

Smith, J. R., Chief Engr., Winrock Farms, Morrilton.
Turner, T. W.*, Vice Pres., Medlin Plumbing & Heating Co. & Mechanical Contractors, Inc., Little Rock.

Louisiana

Butler, L. L., Service Advisor, J. M. Light, Shreveport.

Oklahoma

Palmer, J. E.*, Mech Engr., Palmer Plbg. Htg. & A-C. Co., Tulsa.
Thomas, W. L., Service Engr., Thomas Engineering Co., Oklahoma City.

Texas

Ascherl, J. L.*, Engr., B. Segall, Jr., Austin.
Carpenter, E. W., Chief Design Engr., Dixson-Caperton Co., Inc., Tyler.
Cook, R. L.†, Engr., United Gas Pipe Line Co., Tyler.
Heitzman, Dick, Design Engr., General Electric Co., Tyler.
Howard, W. L., Evaluation Engr., General Electric Co., Tyler.
Ingle, L. A. Jr.*, Appl. and Sales Engr., J. R. Dowdell & Co., San Antonio.
Kaastad, M. N.*, Mgr. Ind. Insulations Cont. & Sales, E. O. "Gene" Wood Co., Fort Worth.
Kelly, W. O., Sales Engr., The Trane Co., Dallas.
Newlin, B. P., Engr., Texas Distributors, Inc., Tyler.
Picht, D. E.*, Engr., Condair Corporation, Dallas.
Ratliff, L. A. Jr., Field Engr., Carrier-Bock Co., Dallas.
Schneider, R. F., Appl. Engr., The Elliott Co., Dallas.
Tye, J. L.*, Vice-Pres. Les Tye Co., Fort Worth.
Walker, M. T.*, Chief Engr., Sales, Torsco, Inc., Fort Worth.

REGION IX

Colorado

Bindner, L. R.*, Sales & Appl. Engr., The Trane Co., Denver.
Deming, H. R.*, Sales Engr., The Trane Co., Denver.
Mullin, J. B., Repr., Clarke-Pacific, Denver.

Kansas

Krogman, F. W., Dist. Repr., Gordon & Platt Burner Co., Shawnee.

Nebraska

Cardella, R. W., Design & Appl. Engr., Natkin & Co., Omaha.

South Dakota

Hayter, K. S., Director of Physical Plant, South Dakota State College, Brookings.

REGION X

Arizona

Dexter, R. C., Sales Engr., Tom Babington, Phoenix.
Timmerman, G. H., Grad. Design Engr., Lowry & Sorensen Engr. Co., Phoenix.

British Columbia

Bruce, R. S., Sales Repr., Fiberglas Canada Ltd., Vancouver.
Racine, R. W.*, Htg. & A-C. Sales Engr., British Columbia Electric, Vancouver.
Wilkes, A. E., Maint. Supvsr., Hudson's Bay Co., Vancouver.

California

Booth, J. B. Jr., Proj. Engr., Keller & Cannon, San Francisco.
Chomik, Steven, Sales Engr., Dust Control, Inc., Hawthorne.
Christian, Scott, Design & Appl. Engr., Jackson & Blanc, San Diego.
Gramaglia, S. L., Estimator, Schlegel Htg. & Vent. Co., Menlo Park.
Greer, J. E., Dist. Mgr., Chrysler Corp., Airtemp Div, Los Angeles.
Lloyd, K. M., Sales Engr., G. J. Jamas Co., (Barber Colman), San Francisco.
Sudtell, H. D., Vice-Pres., J. S. Hamel Eng. Co., Los Angeles.
Will, C. H. Jr., Sr. Process Technologist, Owens-Corning Fiberglas, Santa Clara.

Washington

Gordon, L. W., Draftsman, Culler, Gale, Martell, Norrie & Davis, Spokane.
Robins, C. S., Research Engr. A, Boeing Co., Renton.

FOREIGN

England

Brown, N. W., Sr. Design Engr., Brooks Air & Heat Systems, Ltd., Croydon, Surry.
Dale, K. W., Owner, K. W. Dale & Partners, London.

France

Blondel, P. E.*, Mgr. & Sales Mgr., Ste Ameliorair, Paris.

Italy

Belli, G. C., Executive, Gen. Mgr., E. Origgi, Rome.
Bennello, Franco, Mgr., Breda Finia, Milan.

Kuwait

Mallick, S. A., Tech. Asst., Government of Kuwait.

Scotland

Bowes, W. C., Asst. Design Engr., The Corporation of Glasgow, Glasgow.

South Wales

Arnold, C. J.†, Supt. Engr., Morgannwg Hospital Management Committee, Glamorgan.

STUDENTS

Cummins, R. S., Syracuse University, Syracuse, N. Y.
Grasso, F. D., University of Nebraska, Lincoln, Neb.
Royce, Pelham, University of Houston, Houston, Texas.
Trentham, D. L., Mississippi State University, State College, Miss.
Yee, W. T., University of Wisconsin, Madison, Wisc.

ASHRAE	1962	1963	1964
NATIONAL MEETINGS	Jan. 29-Feb. 1 Semiannual St. Louis, Mo.	Feb. 1-14 Semiannual New York, N. Y.	January 27-29 Semiannual New Orleans, La.
AHEAD	June 25-27 69th Annual Miami Beach, Fla.	June 24-26 70th Annual Milwaukee, Wisc.	June 29-July 1 71st Annual Cleveland, Ohio

ASHRAE INTERSOCIETY REPRESENTATIVES

AMERICAN STANDARDS ASSOCIATION PROJECTS
(Sponsored by ASHRAE)

	Representatives:
A-114 Application Standards for Thermal Insulating Materials	M. W. Keyes, Chairman
B-9 Safety Code for Mechanical Refrigeration Alternates: **W. W. Grear** **A. J. Hess** **W. W. Higham**	**R. L. Williams**, Chairman **J. R. Chamberlain**, Vice Chairman **S. R. Hirsch** **A. I. McFarlan**
B-38 Household Refrigerators and Home and Farm Freezers	**E. C. McCracken**, Chairman **W. W. Higham**
B-53 Refrigeration Terms and Definitions	**G. B. Priester, Chairman** **H. J. Ryan**
B-59 Mechanical Refrigeration Installations on Shipboard	**W. L. Keller**
B-60 Methods of Testing for Rating Thermostatic & Constant Pressure Expansion Valves	**D. C. Albright**
Z-9 Safety Code for Exhaust Systems	**W. M. Wallace, II** Alternate: **W. S. Bondy**
Z-74 Fundamentals of Performance of Effluent Air and Gas Cleaning Equipment	**K. E. Robinson**

AMERICAN STANDARDS ASSOCIATION PROJECTS
(Not Sponsored by ASHRAE)

A-13 Scheme for Identification of Piping Systems	**H. H. Bond** **Crosby Field**
A-40.8 National Plumbing Code	**Fred Janssen**
A-53 Building Code Requirements for Light and Ventilation	**J. G. Eadie**
A-62 Coordination of Dimension of Building Materials & Equipment	
A-119 Mobile Homes and Travel Trailers	**R. G. Dodds** **J. L. Heiman**
B-2 Pipe Threads Representative:	
B-16 Standardization of Pipe Flanges and Fittings	**C. W. Hudzietz**
B-19 Safety Standards for Compressor Systems	
B-31 Code for Pressure Piping	**J. L. Wolf** **S. E. Rottmayer** (only on Subcommittee for revision of Sec. 5 Refrigerant Piping)
B-40 Indicating Pressure and Vacuum Gages	**Bernhard Willach**
B-72 Dimensional Standards for Plastic Pipe	**W. J. Olvany**
B-76 Industrial Cooling Towers	**P. A. Bourquin** **John Engalitcheff, Jr.**
B-78 Heat Exchangers for Chemical Industry Use	**C. E. Drake**
C-85 Terminology for Automatic Controls	**C. H. Burkhardt**
C-96 Temperature Measurement Thermocouples	**W. A. Spofford**
K-61 Storage and Handling of Anhydrous Ammonia and Ammonia Solution	**C. F. Holske**
S-1 Physical Acoustics	**R. E. Parker**
Y-1 Abbreviations	**N. N. Wolpert** Alternate: **C. H. Flink**
Y-10 Letter Symbols	**B. E. Short** Alternate: **C. H. Flink**
Y-14 Drawings and Drafting Room Practice	**F. Honerkamp** **H. J. Donovan**
Y-32 Graphical Symbols and Designations	**E. H. Munier** Alternate: **C. H. Flink**
Z-11 Petroleum Products and Lubrication	**B. L. Evans** **W. J. Simpson**

	Representatives:
Z-17 Preferred Numbers	D. J. Renwick
Z-48 Method for Marketing Portable Compressed Gas Containers to Identify the Material Contained	Herbert Wolf
Z-62 Uniform Industrial Hygiene Standards	A. D. Brandt
Z-84 Glossary of Environmental Terms	C. F. Kayan

AIR-CONDITIONING AND REFRIGERATION INSTITUTE

ASHRAE-ARI Standards Liaison Committee	A. S. Decker H. P. Tinning

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

Subcommittee on the Evaluation of Hermetic Motor Insulation	R. T. Divers

AMERICAN MEAT INSTITUTE

ASHRAE-AMI Meat Packing	K. E. Nielsen, Chairman E. N. Johnson B. C. McKenna C. D. Macy F. P. Neff K. E. Wolcott

AMERICAN PUBLIC HEALTH ASSOCIATION, INC.

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AMERICAN SOCIETY OF AGRICULTURAL ENGINEERS

Plant and Animal Husbandry	M. K. Fahnestock

AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PTC-23 Atmospheric Water Cooling	A. L. Hesselschwerdt, Jr.
PTC-25 Safety and Relief Valves	M. W. Garland

AMERICAN SOCIETY FOR TESTING MATERIALS

	Representatives:
A-5 Corrosion of Iron and Steel	
B-3 Corrosion of Non-Ferrous Metals and Alloys	R. C. McHarness
C-16 Thermal Insulating Materials	C. F. Kayan E. R. Queer
Planning Committee on Thermal Insulating Materials	E. R. Queer
D-2 Petroleum Products and Lubricants	
D-3 Gaseous Fuels	E. A. Norman, Jr.
D-19 Industrial Water	D. R. Walser
D-22 Methods of Atmospheric Sampling and Analysis	R. J. Walker

CANADIAN STANDARDS ASSOCIATION

Safety Code for Hospital Hazards	J. Klassen

INTERNATIONAL ORGANIZATION OF STANDARDIZATON

Advisory Committee to ASA on ISO/TC 86 Refrigeration	A. T. Boggs, III Urban Bowman Leon Buehler, Jr. W. A. Grant J. R. Schreiner T. B. Simon W. E. Ural R. L. Williams

NATIONAL ACADEMY OF SCIENCES—NATIONAL RESEARCH COUNCIL

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NATIONAL ASSOCIATION OF CORROSION ENGINEERS

Intersociety Corrosion Committee	F. N. Speller

NATIONAL FIRE PROTECTION ASSOCIATION

56 Hospital Operating Rooms	R. P. Gaulin N. Glickman
90A-90B Air Conditioning	A. I. McFarlan
91 Blower Systems	W. S. Bondy

BULLETINS AND CATALOGS

Performance Curves. To indicate the performance of a new line of Turbo Blower Equipment, a set of 18 composite performance curves is offered. The equipment, 4 through 24 oz, is direct-connected, 3500 rpm, and is described and illustrated in Bulletin TB-109.
General Blower Company, 8626 Ferris Ave., Morton Grove, Ill.

Freeze-Drying. To meet laboratory freeze-drying requirements, this company offers three units: a Universal Model incorporating a mechanical refrigeration unit in addition to a dry ice and alcohol condenser, a Laboratory Model, designed to use limited space, and a Bench Model, designed for compact size and ease of portability. Descriptive of the line is Bulletin 2345.
American Instrument Company, Inc., 8030 Georgia Ave., Silver Spring, Md.

Finned Tube. Added to this line of heat exchanger and condenser tube products, finned heat transfer tube, integral low fin type, is described in an eight-page bulletin. Dimensional data, specifications, alloys and tempers are covered. Finning is standard for all combinations of diameters and wall thickness at 19 fins per linear in.
Scovill Manufacturing Company, Mill Products Div, 99 Mill St., Waterbury 20, Conn.

STAFF CHANGES

M. A. MAYERS **H. P. TINNING**

Implementing of recent ASHRAE changes brings to the Headquarters Staff Martin A. Mayers as Manager of Research and Herbert P. Tinning as Technical Secretary. An earlier announcement, September JOURNAL page 92, is thus in part counteracted, as Clark Humphreys has joined instead the U. S. Public Health Service in Cincinnati.

Martin A. Mayers, most recently Chief Design Engineer with Sanderson & Porter, Inc., there administered engineering and service sections relating to utility steam power plants. Previously, he was Chief Mechanical Engineer, Special Products, with Burns and Roe, Inc., where as head of the department he was responsible for the mechanical engineering design of aircraft engine test facilities, wind tunnels and rocket launching facilities. Earlier responsibilities have included staff work with M. W. Kellogg Company, Elliott Company and Carnegie Institute of Technology.

Mr. Mayers, a graduate of Yale University, Mechanical Engineering, B.S. (summa cum laude), is a member of ASME and has been Chairman, Power Test Code Subcommittee on Fuels.

Return to the staff of Herbert P. Tinning as Technical Secretary follows an interval during which he was associated successively with Dunham-Bush, Inc., as New Jersey Field Engineer and with Virginia Smelting Company as a Water Chemical Field Engineer. Mr. Tinning will be recalled as former Assistant Secretary – Membership with predecessor ASRE. He is a graduate of Stevens Institute of Technology.

Directional Diffuser. Data on an all-extruded aluminum directional diffuser is provided by twelve-page Catalog D-7-61. Five frame styles and fourteen core patterns are shown and detailed installation information and selection data are provided. Available in square and rectangular designs, which include one-way, two-way opposite, three-way, four-way and two-way corner flow, the Type D diffuser is offered with snap-in, bevelled, drop collar, flange and lay-on frames.
Waterloo Register Company, Inc., P. O. Box 147, Waterloo, Iowa.

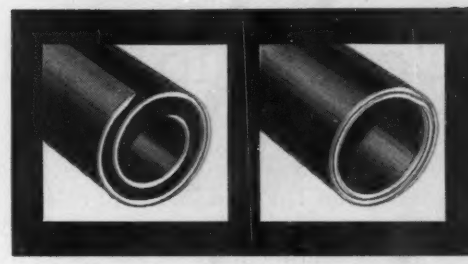
Variable Speed Belts. Tables in 20-page Bulletin 24103 provide information on manufacturers of equipment using variable speed belts and belt numbers of these manufacturers and the corresponding belt number of this company. Belts are listed according to size, using the new standardized numbering system, which conveys width of sheave groove in sixteenths, groove angle in degrees and belt pitch length in inches and tenths.
T. B. Wood's Sons Company, Chambersburg, Pa.

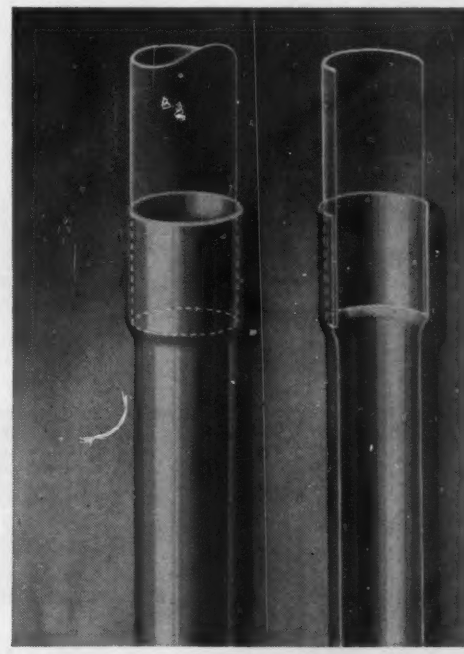
V-Band Couplings. "Economies of the V-Band Coupling", 16-page Bulletin SDP-2, gives specific cost comparisons between V-bands and other joining methods. Adaptability of design is discussed and savings in weight and installation or assembly time are covered.
Aeroquip Corporation, Marman Div, 11214 Exposition Blvd., Los Angeles 64, Calif.

Heat Exchangers. For controlled steam heating of water, this line of heat exchangers is covered in 30-page Catalog AX60. Included are: selection information, rating chart, capacities in gpm and pressure drop in ft of water. A diagram of a typical hookup also is provided. Units described are converters, and radiant heating, snow melting, instantaneous water, booster, swimming pool, storage tower and closed feedwater heaters.
Alstrom Corporation, 790 E. 176th St., New York 60, N. Y.

Centrifugal Pumps. Addition of two new sizes doubles the capacity range of the Model 3305-06 two-stage centrifugal pump line. Now available in sizes 6 x 8-17 and 8 x 10-17, these units provide capacities up to 3000 gpm for heads up to 550 ft at 1750 rpm. Extensive information, including performance curves, material spe-

(*Continued on page* 96)





Carrier to be honored by ASHRAE-ASME

On November 28-29, ASHRAE and the Process Industries Div of ASME will co-sponsor a meeting, luncheon and symposium in New York to honor the pioneering work in psychrometry of Willis H. Carrier and to explore the history, development and present state of art of the psychrometric and cryogenics fields.

This month marks the 50th Anniversary of Willis H. Carrier's exposition of the Rational Psychrometric Formula before a meeting of the ASME (November 1911). The coming meeting serves primarily to commemorate that technologically historic act. A professionally distinguished group of industrial and academic leaders will officiate, speak and present papers of pertinent and timely interest.

The opening session of the meeting is scheduled for Tuesday, November 28 at 9:30 a.m. Eugene Ambrose of the American Electric Power Service Corp. and Leo J. Riconda of General Mills Co. will be Chairman and Vice Chairman, respectively. Papers on Psychrometrics and Modern Comfort, Psychrometrics and Human Disease and Physiological Reactions to Psychrometric Extremes will be presented by Ralph Nevins, Professor and Head, Mechanical Engineering Dept., Kansas State University; Joseph L. Hollander, Assistant Professor of Medicine, University of Pennsylvania; and Lucien L. Brouha, Employee Relations Dept., E. I. duPont de Nemours and Co., authors of the papers, in that order.*

At the luncheon, the same day, a joint presentation of a scroll in observance of Dr. Willis H. Carrier's publication of the "Rational Psychrometric Formula" will be made by ASHRAE President John Everetts, Jr., and ASME President William J. Byrne. Presiding at this meeting will be C. F. Kayan, Professor, Dept. of Mechanical Engineering, Columbia University. Chairman of the Board Cloud Wampler of the Carrier Corp. will speak on The Legacy of Willis Carrier.

The afternoon session (2:30 p.m.) will be made up of a symposium on psychrometrics and the history and development of psychrometric charts. A new psychrometric chart adopted recently by ASHRAE will be described. This session will be under the chairmanship of P. B. Gordon of Wolf & Munier; Vice Chairman, Royal S. Buchanan, American Motors Corp.

* Papers for this session are ASHRAE papers.

Amana Refrigeration, Inc., Director of Purchasing, Mr. A. C. Schmieder, is shown with coils whose rigidity and resistance to damage is credited to Anaconda Aluminum tempered fin stock.

There's a lot more to buying materials than checking quality and delivery—although Amana Refrigeration, Inc., places plenty of emphasis on both. Here's the story of some extra values Amana gets because, from the inception of their cooling coil production program, Anaconda Aluminum was contacted and permitted to work closely with them.

When the idea was still new, Amana and Anaconda Aluminum worked together to develop use of no-cost fiber cores to protect fin stock. In the same manner, Anaconda Aluminum and Amana developed ways to utilize new alloys. And as Amana's sales grew—and along with this growth the need for coils grew—Anaconda Aluminum also worked with Amana's die suppliers to assure maximum efficiency.

Amana took advantage of Anaconda Aluminum engineers' understanding of metallurgical problems—resulting in a switch to a special temper. Production could be made more efficient, handling problems reduced, and fin rigidity increased. In addition, production problems with heavy gauge material in the refrigeration and freezer department were also worked out as the result of this thoroughgoing partnership between Anaconda Aluminum and its customer.

Providing consistent production help, metallurgical assistance, and extra fast delivery when customers need it is part of the product from Anaconda Aluminum. You get proof of it every time you talk with an Anaconda Aluminum customer. For information, contact your Anaconda Aluminum office or write Dept. AJ-11, Anaconda Aluminum Company, Louisville 1, Kentucky.

ANACONDA ALUMINUM

The morning session of Wednesday, November 29, will concentrate on air conditioning and will begin at 9:30. Chairman will be G. C. Beck of the Brooklyn Union Gas Co.; Vice Chairman, Frederic A. Lang, E. I. duPont de Nemours and Co. Papers presented will be: The Economic Application of Gas Turbines to Large-Tonnage Air Conditioning Plants, C. R. Apitz, Clark Brothers Co.; Absorption Air Conditioning, C. Owen Kuhen, Carrier Corp.; Application of Natural Gas Engines to Air Conditioning and Refrigeration, Robert A. D'Amour, Waukesha Motor Co.

A panel discussion on The Rapid Growth of Cryogenics by John C. Fisher of the General Electric Research Laboratory, Donald R. Young of IBM Research Laboratory, Arthur A. Fowle of Arthur D. Little, Inc., and John Macinko of the National Bureau of Standards will mark the final session (2:30 p.m.). Progress of the past five or six years in the application of low-temperature phenomena prompts this survey. The panel will review briefly the subjects of superconductivity, superconducting devices, refrigeration and instrumentation, insulation materials and design principles, following which there will be an informal discussion.

BULLETINS

(*Continued from page* 90)

cifications, interchangeability charts and dimensions, is provided in eight-page Bulletin 722.6.
Goulds Pumps, Inc., 227 Black Brook Rd., Seneca Falls, N. Y.

Hot Water Heating Equipment. Introduced in 16-page Bulletin 1551B are expansion tank, steam and water converters with typical converter installation diagram, plus modifications of other products. Discussed are circulators, valves, fittings, specialties and converters. Helpful to engineers are concise specification details, performance curves, installation diagrams, pressure drop tables and general product information.
Dunham-Bush, Inc., 179 South St., West Hartford 10, Conn.

Air Movers. Descriptive of propeller air movers, an eight-page bulletin includes construction features, dimensional data, electrical characteristics and performance charts. Accessories are illustrated. Information is provided for belted SB air movers, the direct-drive SD and the Economy line.
Greenheck Fan & Ventilator Corporation, Schofield, Wisc.

Two-Stage Seal. Pallseal, a new stainless steel and Viton "A" seal for fluid systems operating between –65 and 500 F and up to 10,000 psi, is the subject of Flyer E15. Describing the construction of these composite V-rings, the bulletin gives technical data on materials, dimensions and standard sizes available.

A soft seat for leak-tight sealing is provided by the Viton "A" primary stage, while the outer shell of the unit maintains the seal under extreme temperatures and pressures, additionally providing a metal-to-metal secondary seal.
Pall Corporation, 30 Sea Cliff Ave., Glen Cove, N. Y.